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A SURVEY AND ECONOMIC ANALYSIS OF ALTERNATE METHODS FOR COOLING CONDENSER DISCHARGE WATER IN THERMAL POWER PLANTS. TASK I REPORT. SURVEY OF LARGE-SCALE HEAT REJECTION EQUIPMENT

John H. Carey, et al

Dynatech R/D Company

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Federal Water Pollution Control Administration
U.S. Department of the Interior
Corvallis, Oregon 97330

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Under

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Section 1

INTRODUCTION

This report presents the results of Task I, Phase I of an overall program to determine the technological and economic consequences of alternative methods of cooling condenser discharge water from thermal power plants.

1.1 Scope of Task I

The first task of this study, which is reported on in this document, is concerned with various schemes for heat rejection on a large scale. A large part of the problem in this area is attributable simply to the sheer magnitude of the quantities of heat which must be rejected. In "order-of-magnitude" terms, a 500 megawatt plant with an efficiency of ~25% must reject approximately five billion Btu/hr (5.0×10^9) of heat to its surroundings. In terms used in conventional cooling equipment, this represents 400,000 tons of cooling (1 ton = 12,000 Btu/hr). (For purposes of comparison, the capacity of the air-conditioning system in Boston's Prudential Center, a fifty-two story office building, is approximately 6,000 tons.) Clearly, the difficulties of extrapolating performance data on existing equipment are severe. In this task, information is presented to identify present state-of-the-art methods of large scale heat rejection and to determine their applicability to this problem.

1.2 Format of Task I Report

This report is divided into six sections plus a bibliography. This first section serves both to define the scope of this task and where it fits into the overall program goals and also to provide a "road-map" through the report itself. The following paragraphs give a brief synopsis of each of the remaining sections.

1.2.1 Section 2 - Thermodynamic Aspects of the Steam Power Cycle

In Section 2, a brief review of the thermodynamics of power generation is presented. This is not intended to be comprehensive nor to deal with the practical aspects of real systems. It serves simply to provide some understanding of how the heat rejection requirement affects the total power cycle, and to set some fundamental thermodynamic limits on the nature of the solution. It will further define much of the nomenclature for the remainder of the discussion. A detailed treatment of the practical aspects of power generation will be done in Task II.

1.2.2 Section 3 - Heat Rejection Equipment - Technical Considerations

In Section 3, information is presented on all heat rejection schemes which may be considered candidates for use with large power plants. These include once-through cooling, cooling ponds, spray ponds, wet cooling towers, dry cooling towers, evaporative condensers, air-cooled condensers, and some advanced concepts. The material presented includes sizing procedures, capital and operating cost data, maintenance, and secondary pollution considerations.

1.2.3 Section 4 - Heat Rejection Equipment - Operating Considerations

Section 4 is organized in the same way as Section 3 and concerns the same types of equipment. It deals, however, with the operating considerations such as theory of operation at design, off-design operation, running procedures, and any unusual characteristics affecting the interface of the cooling system with the power plant.

1.2.4 Section 5 - Methods of Comparison and Selection

In Section 5, the material describing the alternative cooling methods is integrated and criteria are developed for the selection of a particular unit in terms of geographic location, capacity requirements, and economic factors.

1.2.5 Section 6 - Potential Extensions of the State-of-the-Art

In Section 6, some preliminary thoughts are presented to serve as a basis for the work of Task II, Phase II which will deal with advanced concepts in large-scale heat rejection equipment.

Section 2

THERMODYNAMIC ASPECTS OF THE STEAM POWER CYCLE

2.1 Basic Steam Cycle

A block diagram of the basic Rankine steam power cycle is presented in Figure 2.1. All existing steam power plants are variations of this basic cycle. A particular plant may have a number of turbines with steam being reheated between turbines or it may use a multiple stage turbine with a portion of the steam being withdrawn between stages to preheat the water before it re-enters the boiler; there are many variations which may be employed to increase the cycle efficiency above that of the basic steam cycle. However, the performance of the basic Rankine cycle is representative of all steam power systems.

Heat is transferred to the boiling fluid at a constant pressure either from a burning combustible fuel or from a thermonuclear reaction, in order to provide steam at a high pressure and temperature at location ①. The steam is then expanded through a turbine to generate the output power from the cycle. The turbine is generally connected to an electrical generator to provide electrical power. At location ② the steam has a lower temperature and a much lower pressure. In the condenser, heat is rejected from the cycle to an external heat sink at a constant pressure, causing the steam to condense. Thus, all the steam has been converted to water at location ③ . Between locations ③ and ④ a pump raises the pressure of the water to that of the boiler and moves the water into the boiler to complete the cycle.

An excellent way to show the thermodynamic processes occurring in the basic Rankine cycle is the temperature-entropy diagram of the working fluid. (See Figure 2.2.)

The solid lines in Figure 2.2 indicate the basic Rankine cycle. The pumping (3-4) and the expansion in the turbine (1-2) are shown as reversible adiabatic processes for simplicity. It is conventional to install a superheater between the boiler and the turbine. This moves the point (1) into the superheat region point 1' so that the end of the expansion is now 2', a point of high quality. Some water droplets are present but in tolerable amounts. The basic power plant Rankine cycle is shown by the points 3-4-1'-2'-3 in Figure 2.2.

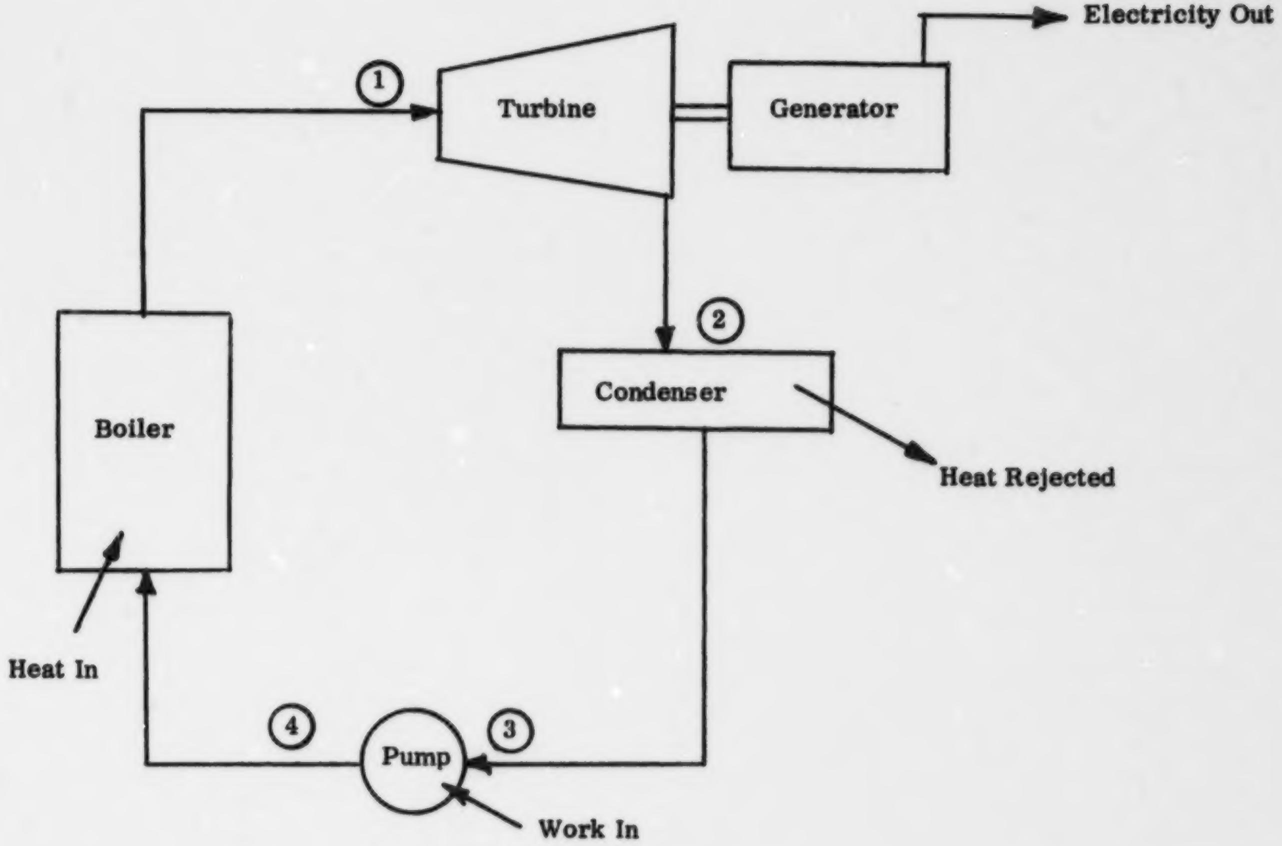


Figure 2.1. Schematic of Basic Rankine Power Cycle

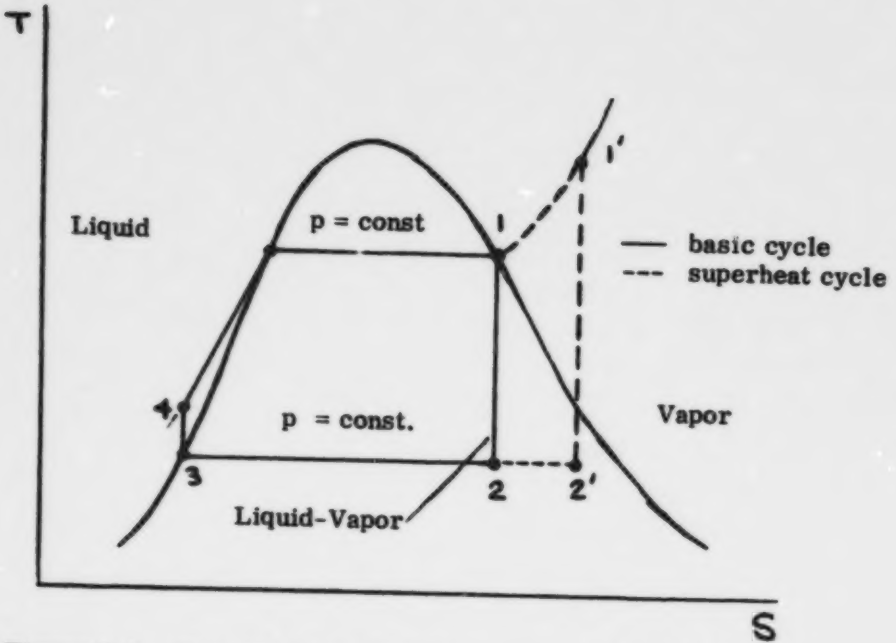


Figure 2.2. Temperature-Entropy Diagrams of Rankine Cycles

2.2 Thermodynamic Efficiency (η_{th})

The efficiency of a steam power cycle is defined as the net work output divided by the heat input to the system:

$$\eta_{th} = \frac{W_{turbine} - W_{pump}}{Q_{IN}} \tag{2-1}$$

Conservation of energy requires that the equation below be satisfied

$$Q_{IN} + W_{pump} - Q_{rej} - W_{turbine} - P_{Loss} \text{ (Plant Losses)} = 0 \tag{2-2}$$

Equations (2-1) and (2-2) may be combined and written as follows;

$$Q_{rej} = \left(\frac{1}{\eta_{th}} - 1\right) W_{turbine} - \left(\frac{1}{\eta_{th}} - 1\right) W_{pump} - P_{Loss} \tag{2-3}$$

The second term in Equation (2-3) is very small, since the pump power represents a negligible portion of the turbine power. Therefore, the balance between turbine power and heat rejected may be approximated by the following relation:

$$Q_{rej} = \left(\frac{1}{\eta_{th}} - 1\right) W_{turbine} - P_L \qquad (2\text{-}4)$$

For a plant which must generate a given amount of power, it is clear that any decrease in thermodynamic efficiency is accompanied by an increase in rejected heat.

The thermodynamic efficiency can also be written in terms of the thermodynamic properties.

Q_{IN} - heat added in the steam generator

$$Q_{IN} = \dot{m}\,(h_{1'} - h_4)$$

$W_{turbine}$ - work extracted from the turbine

$$W_t = \dot{m}\,(h_{1'} - h_{2'})$$

W_{pump} - work required to pump the liquid

$$W_p = \dot{m}\,(h_4 - h_3)$$

Q_{rej} - heat required to condense the exit steam from the turbine

$$Q_{rej} = \dot{m}\,(h_{2'} - h_3)$$

where $\dot{m}$ = flow rate of water through the system

h_i = enthalpy of the water at various stations.

Thus, the expression for efficiency can be rewritten as follows:

$$\eta_{th} = \frac{(h_{1'} - h_{2'}) - (h_4 - h_3)}{(h_{1'} - h_4)} \qquad (2\text{-}5)$$

or

$$\eta_{th} = \frac{(h_{1'} - h_4) - (h_{2'} - h_3)}{(h_{1'} - h_4)} \qquad (2\text{-}5a)$$

2.3 Methods of Decreasing Heat Rejection (Increasing η_{th})

Any decrease in the amount of heat rejected resulting from internal power plant characteristics is obtained from altering the thermodynamic power cycle. The main area of concern to power plant designs in the past has been increasing the thermal efficiency, since this change decreases the cost per kwh. The present state of the art in power plant design indicates that sizable gains in efficiency are a thing of the past.

In this section, methods currently used in power plants to increase the thermal efficiency (thus to decrease Q_{rej}) as shown in Equation (2-4) are discussed and equations for computing η_{th} are presented.

2.3.1 Lowering the Condenser Pressure

The thermal efficiency of a plant is largely controlled by the pressure at which condensation occurs in the condenser. In a two-phase mixture, only one thermodynamic property is independent. Usually the temperature or the pressure is specified and the other properties are determined by the saturation line of the fluid. As a result, when the temperature of the condenser is decreased, the corresponding pressure will also decrease. The highest cycle efficiency is obtained by operating with the condenser pressure as low as possible. Calculations of thermal efficiency and heat rejection rate have been performed for a hypothetical steam power plant which generates power at the rate of 120 Mw. The boiler provides steam at a pressure of 1500 lbf/in^2 and a temperature of 1000°F and the turbine efficiency is 80%. Figure 2.3 illustrates the values of thermodynamic efficiency and heat rejection rate for this plant using a range of values of condenser pressure and temperature while maintaining a constant power output.

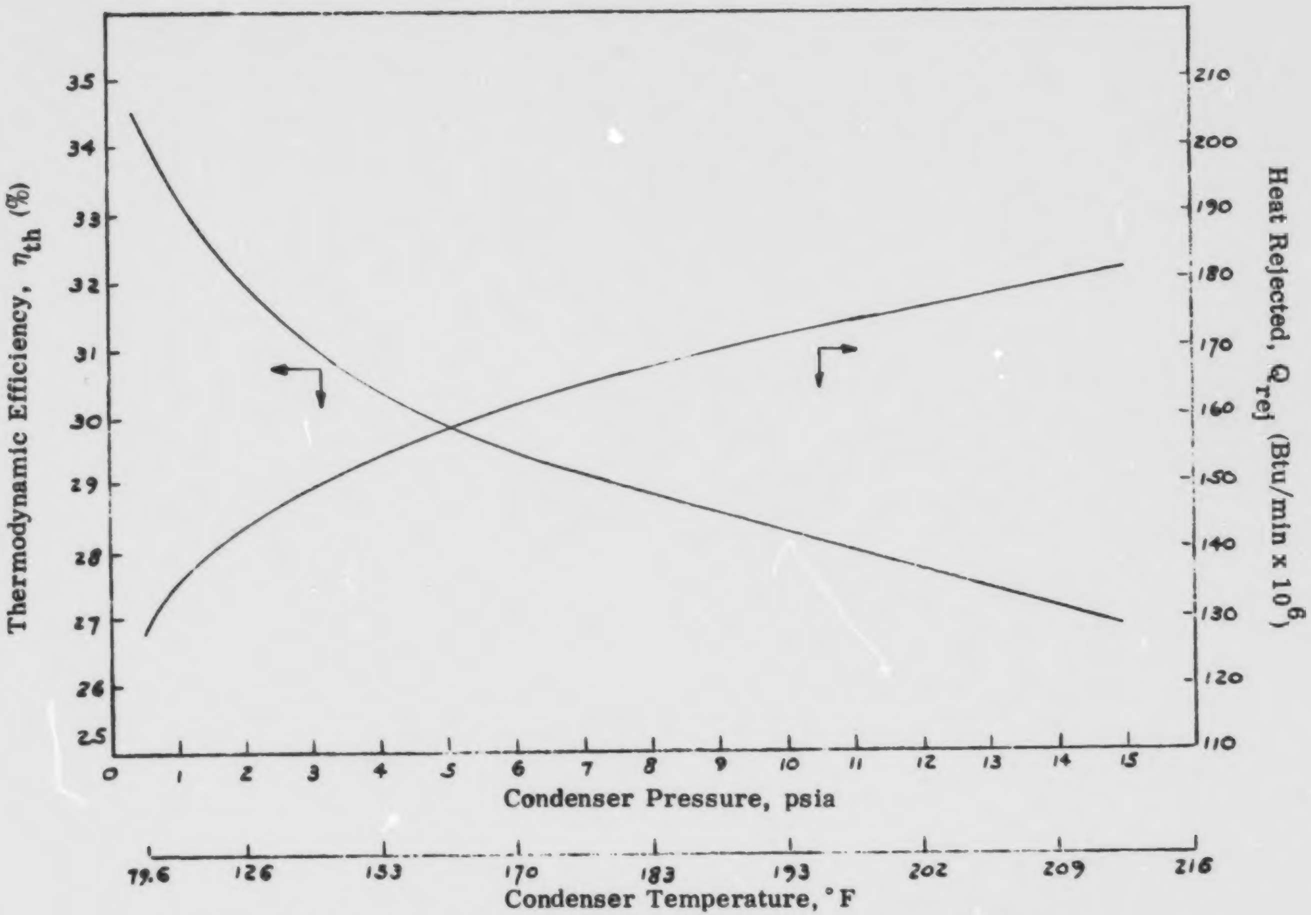


Figure 2.3. Thermodynamic Efficiency and Heat Rejection Rate As A Function of Condenser Pressure

The effect of condenser temperature variations upon the condenser heat rejection rate is clear from Figure 2.3. The condenser temperature is determined by the rate at which heat can be removed from the condenser to a heat sink or cooling source. The method presently employed in many power plants is to draw water from a nearby river and use this to cool the condenser coils, thereby transferring heat from the condenser to the diverted river water. Regardless of the method chosen to remove heat from the condenser, the ability to reject this heat is an essential part of the steam power cycle and is a vital factor in determining the overall efficiency of the power plants as well as the amount of heat rejected and the temperature at which it is transferred to the heat sink.

2.3.2 Increasing the Boiler Pressure

In Figure 2.4 a Rankine cycle is shown with two upper pressure levels p_1 and p_2.

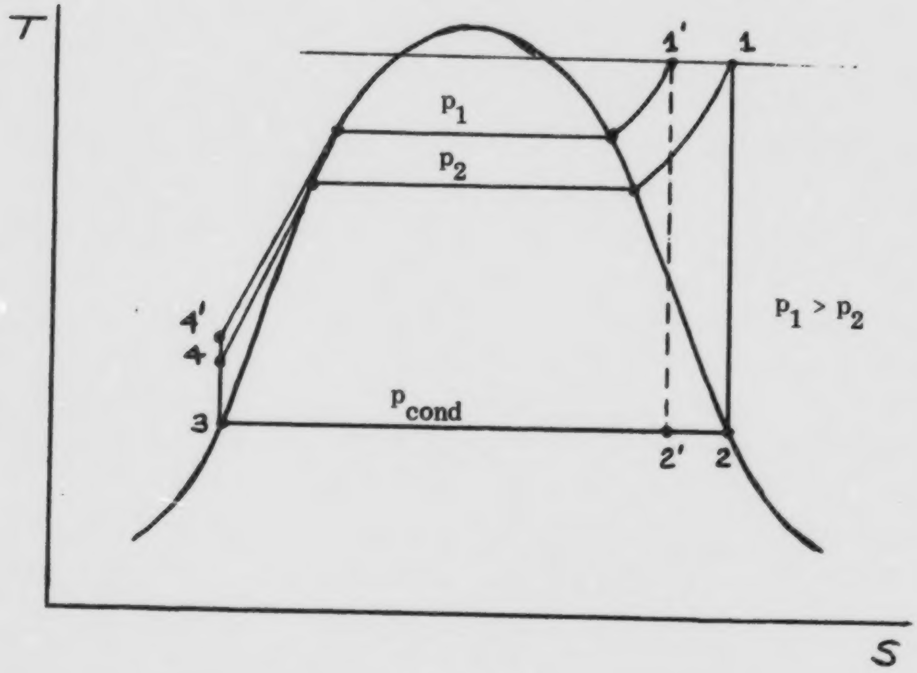


Figure 2.4. Effect of Increased Boiler Pressure on Power Cycle

The efficiency of a cycle operating at p_2 is:

$$\eta_{th} = \frac{(h_1 - h_2) - (h_4 - h_3)}{(h_1 - h_4)} \qquad (2\text{-}7)$$

at p_1

$$\eta_{th} = \frac{(h_{1'} - h_{2'}) - (h_{4'} - h_3)}{(h_{1'} - h_{4'})} \qquad (2\text{-}8)$$

The numerator (net work) of both equations remains approximately constant, while the denominator (the heat added at the steam generator) is smaller at the higher pressure p_1. Thus at constant power the amount of heat rejected at the condenser would decrease as the pressure increased.

2.3.3 Increasing the Amount of Superheat

An increase in the superheat in the boiler, increases the work obtainable from the turbine while it also increases the heat added in the boiler. However the percentage increase in net work is greater than the percentage increase in additional heat required since the heat is being added at a higher average temperature. Thus the overall effect is an increase in the efficiency of the cycle. At constant power level, this results in less heat rejected.

2.3.4 Reheat Cycle

The reheat cycle has been developed to take advantage of the increased efficiency with higher pressures, and yet avoid excessive moisture in the low pressure stages of the turbine, without going to excessive turbine inlet temperatures.

In this cycle steam is extracted from the turbine after being partially expanded and is reheated in the boiler before it is expanded further in the turbine. This cycle is shown on a T-S diagram with solid lines in Figure 2.5

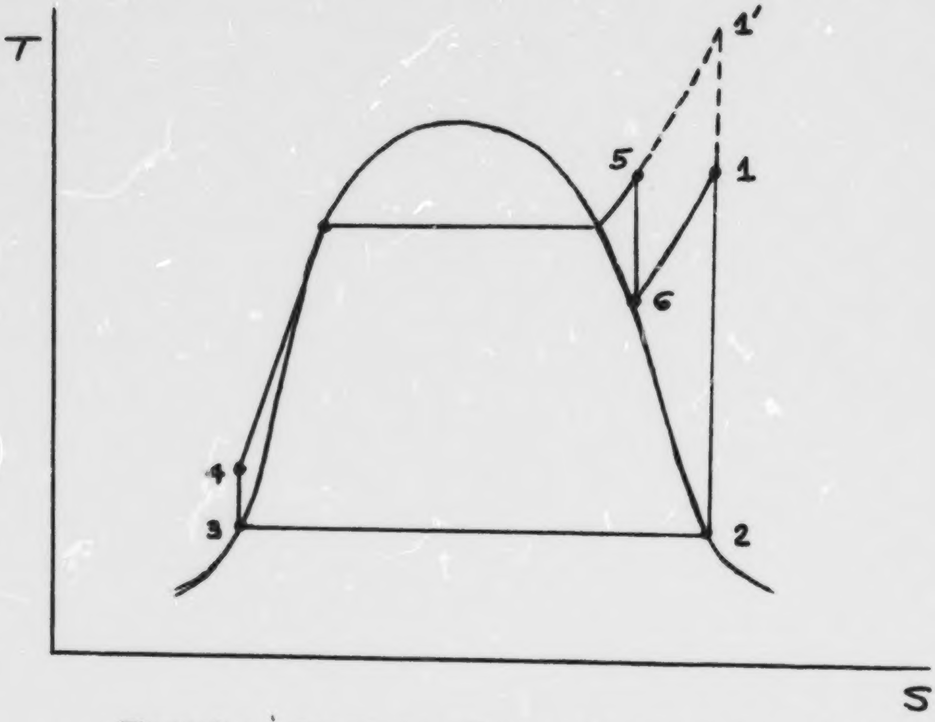


Figure 2.5. Temperature-Entropy Diagram of Reheat Cycle

The thermal efficiency of this process is given by:

$$\eta_{th} = \frac{(h_5 - h_6) + (h_1 - h_2) - (h_4 - h_3)}{(h_5 - h_4) + (h_1 - h_6)} \tag{2-9}$$

This modification is not a primary improvement on the cycle. If superheat were added up to point 1', as shown in Figure 2.5, and no reheating was done, the thermal efficiency for the simple cycle (dashed lines) would be greater than that indicated by Equation (2-9). The reason the reheat cycle is used is to overcome equipment limitations which would not allow operation at temperatures existing at point 1'.

2.3.5 Regenerative Cycle

This is a modification to the Rankine cycle which extracts steam from some intermediate point in the turbine and heats the water entering the boiler thus decreasing the amount of heat added at the boiler. The net work of the cycle is also decreased since all the steam is not expanded in the turbine. However, since the average temperature at which heat is added to the system from outside is increased, the net effect is an increase in thermal efficiency of the cycle and a subsequent reduction of the heat rejected at the condenser. There are two types of feedwater heaters: open and closed. The open feedwater heater has the steam from the turbine and the liquid from the condenser in actual contact with each other. The closed feedwater heater keeps the two fluids separate while the transfer of heat takes place.

Open feedwater heaters have the advantage of being less expensive coupled with the disadvantage of requiring a pump to handle the feedwater between each heater.

Figure 2.6 shows a schematic of a cycle with one open feedwater heater and Figure 2.7 shows the accompanying T-S diagram for the cycle.

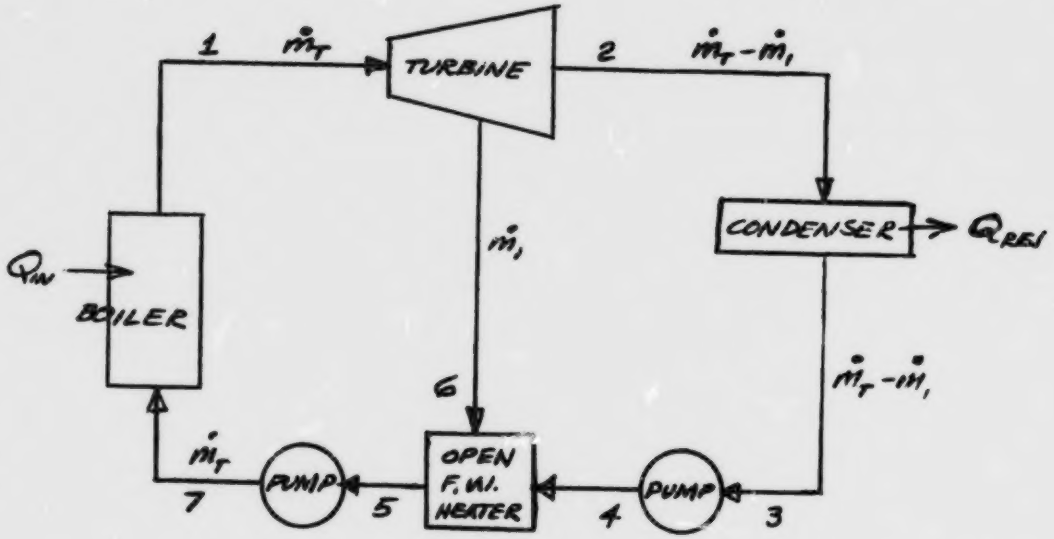


Figure 2.6. Schematic of Regenerative Cycle

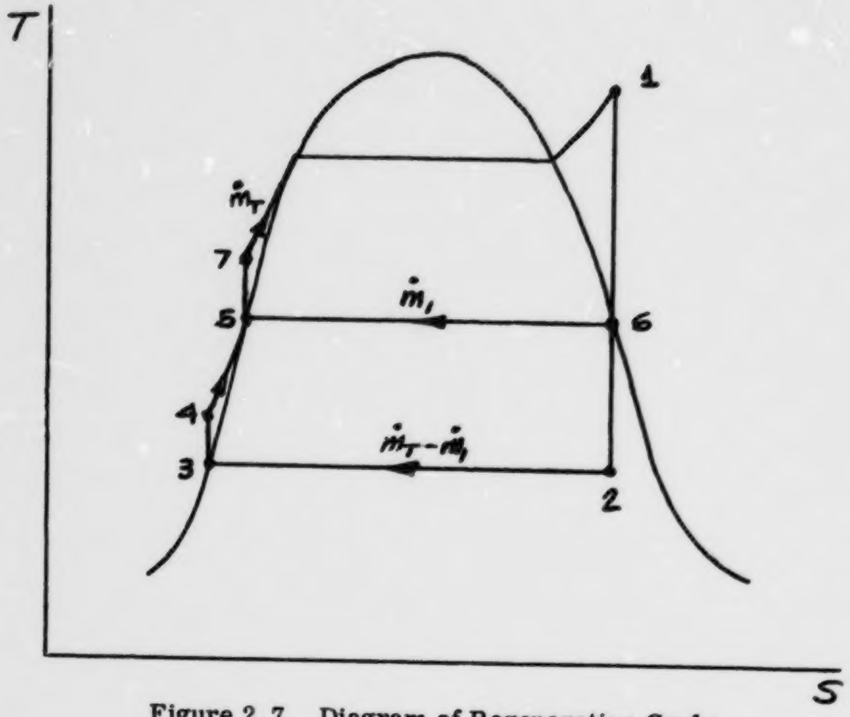


Figure 2.7. Diagram of Regenerative Cycle

The expression for the thermal efficiency for this system is:

$$\eta_{th} = \frac{(h_1 - h_6) + \dfrac{m_T - m_1}{m_T}(h_6 - h_2) - (h_7 - h_5) - \left(\dfrac{m_T - m_1}{m_T}\right)(h_4 - h_3)}{(h_1 - h_7)}$$

where m_T = total mass flow rate

m_1 = mass flow extracted from the turbine

2.3.6 Supercritical and Binary Cycles

Two other cycle modifications have been used in modern power plants; supercritical steam generation and binary cycles (mercury-steam). They have not been applied to large nuclear installations for the following reasons. The pressures and temperatures encountered in a supercritical cycle (1000° F and 3200 psia) exceed those permitted in a water cooled reactor. Typically, conditions in nuclear plants run about 600° F and 1000 psia for boiling water reactors and 600° F and 2000 psia for pressurized water reactors.

The binary-vapor cycle has been used as a "topping unit" in small plants in an attempt to overcome some undesirable thermodynamic characteristics of water: (1) the high specific heat of the liquid phase; (2) low latent heat of evaporation at high pressures, and (3) high boiler pressures. However, the differences in latent heat are such that the mass flow rate of mercury must be 7 - 10 times that of the steam. The result is that the properties which determine the plant size and cost outweigh the thermodynamic attractiveness.

REFERENCES - Section 2

The following references contain general treatments of the thermodynamic aspects of steam power cycles. Task 2 will examine the design and operation of power plants in detail.

2.1 Sorenson, Harry A., Principles of Thermodynamics, Holt Rinehart and Winston, New York, 1961.

2.2 Keenan, Joseph H., Thermodynamics, John Wiley & Sons, Inc., New York, 1941.

2.3 Mooney, David A., Mechanical Engineering Thermodynamics, Prentice-Hall, Inc., Englewood Cliffs, New Jersey, 1953.

Section 3

HEAT REJECTION EQUIPMENT—TECHNICAL CONSIDERATIONS

3.1 Introduction

It has been established in Section 2 that heat rejection at the low temperature of the steam power cycle is necessary to its operation. To transfer this heat from the condenser a driving potential either in the form of temperature or enthalpy (for evaporative systems) must be present. If the immediate sink is of finite size, it will increase in temperature until heat can no longer be transferred unless an equal amount of heat is also removed from the sink. The rejected heat is thus transferred from one sink to another. From this point of view, deep space is the ultimate heat sink. For example, in run-of-river cooling, the rejected heat follows a path from the condenser through the river water to the atmosphere and is then radiated to deep space. The temperature of the river and/or the atmosphere may increase in the process.

Present heat rejection systems transfer the heat from the condensing steam to either water which is eventually cooled in the atmosphere or directly to the air.

The heat may be transferred by two modes; latent and sensible. Latent heat transfer, or more commonly evaporation, occurs at constant temperature and involves the amount of heat (energy) used in transforming the water from liquid to the vapor state. The amount of energy is called the latent heat of vaporization and is taken from the remaining water, lowering its temperature. Although evaporation is usually called heat transfer, it is actually a removal of energy by mass transfer. The difference between the partial pressure of water in the air and the vapor pressure of the water is the driving potential.

The sensible heat transfer occurs due to a difference in water temperature and air dry-bulb temperature. As sensible heat is transferred, the air heats up and the water cools down. Often, both modes operate simultaneously with approximately 75% of the total heat transferred by evaporation and the remaining 25% by sensible heat transfer in cooling towers, for example.

The following heat rejection systems have been considered as candidates for use with large power generation systems:

- Once-through cooling
- Cooling ponds
- Spray ponds
- Cooling towers (wet and dry)
- Evaporative condensers
- Air cooled condensers

A discussion of several advanced systems not yet in use has also been included.

3.2 Once-Through Cooling

Once-through cooling is the most common method for removal of heat from power plant condensers in the United States. In this system cool water is diverted from a river or estuary and pumped through the condenser tubes where it picks up the latent heat of the steam condensing on the outer surface of the tubes. The warmed water is then discharged to the river at a point where recirculation of the warm water will not occur. No attempt at cooling the warm water prior to discharge is made in contrast to the other systems considered in this report.

Once-through cooling is the least expensive method, provided a suitable site can be found. Capital and installation costs are usually low although the sea water installations require a slightly more expensive condenser and longer piping. Operating and maintenance costs are also low. However, various forms of aquatic life are forced through the condenser tubes often resulting in high mortality rates and the discharged water is 10 to 20 degrees warmer than the inlet. This sudden change in temperature upsets the ecology of the water lowering the oxygen content, increasing the metabolic rate of the fish and enhancing growth of undesirable bacteria. Many miles are required before the water temperature returns to normal due to the very slow cooling process.

The temperature increase of the cooling water through the condenser may be calculated from:

$$\Delta T = \frac{\text{heat rejected to water, (Btu/hr)}}{\text{water flow rate through condenser, }(\text{lb}_m/\text{hr})}$$

The specific heat of water is 1.0 Btu/$\mathrm{lb_m}$-°F and has been omitted in the equation. This warm water is then discharged to a region of cool water. If the warm and cool waters mix thoroughly, the temperature will decrease quickly to a mixed temperature given by

$$\Delta T_{\mathrm{mixed}} = \Delta T\,p + T_{\mathrm{cool}}$$

p is the percentage of total water flow pumped through the condenser.

On the other hand, thermal stratification may occur, that is, the warm water may simply form a layer on top of the denser cool water and very little mixing between the two layers will occur. Criteria for the establishment and stability of thermally stratified flows have not been fully determined, but the placement of the discharge, the flow patterns of the stream or estuary which govern the advection, and the local fluid velocities and channel characteristics which govern the turbulent mixing of the layers seem to be the more important parameters. The shape and extent of the "thermal plume" often observed at the discharge of plants located on estuaries can be greatly altered by the tidal stage. Movements of the plume upstream or spreading out across the stream followed later by a return to a plume that hugs the bank near the discharge have been observed. Almost complete mixing due to wave action is observed when the warm water is discharged to a bay.

Once the warm water has been discharged, the heat must be dissipated so that the temperatures will return to "normal". The net rate at which heat is transferred across the water surface is determinated by the algebraic sum of seven quantities:

1. Short-wave solar radiation directly from the sun.
2. Long-wave atmospheric radiation primarily from water vapor in the upper atmosphere.
3. Reflected solar radiation.
4. Reflected atmospheric radiation.
5. Long-wave back radiation from the water to the atmosphere.

6. Evaporation of water into the air above the water; and

7. Conduction from the water to the air and surrounding cooler regions of water and land.

A complete description of these phenomena is given in Reference 3.1.

Neglecting the conduction to the surrounding water and land, we can write

$$\frac{dH}{dt} = -K\,(T_W - T_E)$$

where dH/dt = net rate of heat exchange per unit area.

K = thermal exchange coefficient.

T_W = surface temperature of the water.

T_E = equilibrium temperature.

The equilibrium temperature, T_E, is the water temperature at which there is no net exchange of energy between the water and its surroundings. Determination of T_E involves an iteration where one guesses a value of T_E, calculates several functions which are dependent on T_E, and then calculates a new value of T_E. The process is complete when the initial and calculated values of T_E lie within a certain range. A thorough discussion of the method and tables of the required functions is given in Reference 3.1

The energy exchange coefficient K is a function of wind speed, evaporation rate, and equilibrium temperature. The following equation from Reference 3.1 may be used

$$K = 15.7 + (0.26 + \beta)\,bW$$

where W = wind speed

b = empirical evaporation coefficient

β = de/dT (Figure 3.1, Ref. 3.1)

e = saturation vapor pressure

Solutions of the heat transfer equation yielding temperature distributions are obtainable only after simplifying assumptions are made. Edinger and Geyer (Ref. 3.1) present an expression for the exponential decay in temperature downstream of a point where the discharge and stream are completely mixed. This analysis is rather simple in that it assumes the heated layer is completely mixed at a given downstream position; that is, there are no horizontal or vertical temperature gradients present. As heat is carried away from the stream, the longitudinal temperature decays according to

$$T = T_E + \left[\frac{Q_P T_P + (Q_R - Q_P) T_R}{Q_R} - T_E \right] e^{-(K/\rho C_p u d_m) x}$$

where Q_P = plant cooling water discharge rate

Q_R = total flow rate of stream

T_P = plant discharge temperature

T_R = river temperature prior to mixing

u = average stream velocity

d_m = average depth of heated water

ρ = density of water

C_p = specific heat of water

More complicated models than the one presented above have been formulated. However, solution is generally performed numerically using a computer. Typically, a large number of cases must be run to study the effect of time of day, meteorological conditions, and condition of the body of water at different times of the year. The results are not, in general, applicable to a large number of cases.

Capital costs for once-through cooling generally include the condenser, circulating water pumps, screen house, and associated piping. The pumps and piping sizes are determined by the volume of water that must be passed through the condenser. The amount of heat removed by the cooling water is equal to

$$H = \dot{m}\, C_p\, \Delta T$$

where $\dot{m}$ = mass flow rate of cooling water, (lb_m/hr)

C_p = specific heat of water, (1.0 Btu/lb_m/°F)

ΔT = temperature rise of cooling water through the condenser, (°F).

A condenser may be designed to raise a small amount of water a relatively large ΔT or a large amount of water a small ΔT. In the past the availability of water at the plant site in conjunction with economic studies has determined which route would be followed. Pumping through a large amount of water and raising it just a few degrees allows a smaller condenser to be used but the pumping costs are higher.

In addition to the temperature rise and amount of heat to be transferred, the condenser size is determined by the inlet water temperature, the temperature of the condensing steam and the overall heat transfer coefficient. Cost of the condenser will be raised by requiring special construction features and/or special materials (marine use).

The cost of single pass shell and tube condensers as a function of required tube area is given in Figure 3.1 (Ref. 3.4). The area may be computed from

$$H = U_o A\, \Delta T$$

where H = heat removed from condensing steam, (Btu/hr)

U_o = overall heat transfer coefficient, (approximately 500 Btu/hr-ft^2-°F)

ΔT = log mean temperature difference between steam and cooling water, (°F).

The following values are typical for power plants:

H = 3840 Btu/kwh for a fossil fuel plant with 40% thermal efficiency

H = 6410 Btu/kwh for a nuclear plant with 33% thermal efficiency.

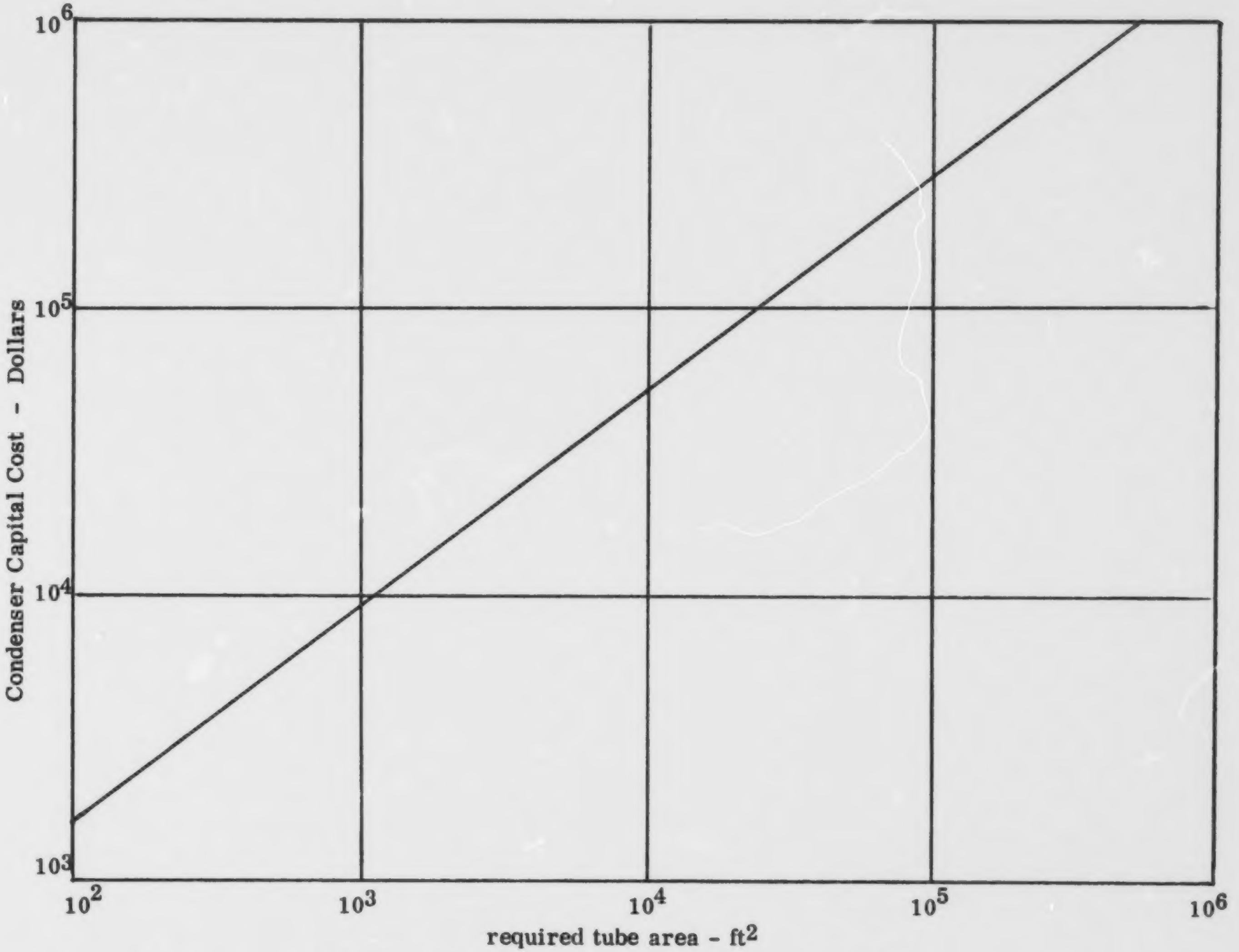


Figure 3.1. Condenser Capital Cost as a Function of Tube Area

A study by Shade and Smith (Ref. 3.2) run-of-river cooling on an 1800 Mw fossil fueled plant with 55° F inlet water temperature, 17° F temperature rise through the condenser and 1.2 inches Hg condenser pressure yielded a cost of $5.00/kw for the installed condenser, pumps, and piping. The condenser for a hypothetical 1000 Mw nuclear plant with an efficiency of 33%, 90° F condensing temperature, 65° F inlet water and 15° F temperature rise through the condenser would cost $1.32/kw using Figure 3.1. Adding this to the $0.84/kw cost for the pumps and doubling the total to account for installation and then adding the $1.00/kw for the intake crib (Steur, Ref. 3.55) yields $5.32/kw. This cost would be slightly higher for a smaller unit, other factors being equal, since specific machinery costs decrease as unit size goes up.

The pump cost was based on a figure of $1.00/gpm. A nuclear plant typically uses 0.84 gpm/kw while a fossil plant uses about 0.42 gpm/kw. Kovats (Ref. 3.3) outlines some of the aspects of water pump selection and specification. Consideration of small, multiple pump designs versus large, single units is made.

Condensers for marine installations will cost about 25% more than conventional units judging from data in Reference 3.4. If we use this figure and identical installation costs and assume $0.50/kw for additional piping, the cooling system cost rises to $6.15/kw for our hypothetical 1000 Mw nuclear plant. Shade and Smith give a figure of $6.00/kw for a bay or lake cooling scheme where the additional cost is due to longer piping. Eicher (Ref. 3.56) lists the additional cost of a once-through marine installation at $1.00/kw.

In summary, once through cooling of large power plants ranges from $5.00/kw to $6.50/kw. Run-of-river cooling will generally provide the most economical operation while installations requiring the use of sea water and long runs of piping will be the most expensive. It is clear that although simple run-of-river cooling is quite economical, considerations of available sites and stream temperature increases will virtually eliminate this method as a heat rejection scheme.

3.3 Cooling Ponds

A cooling pond is a body of water into which the warm condenser discharge is pumped so that it may be cooled and eventually recirculated through the condenser.

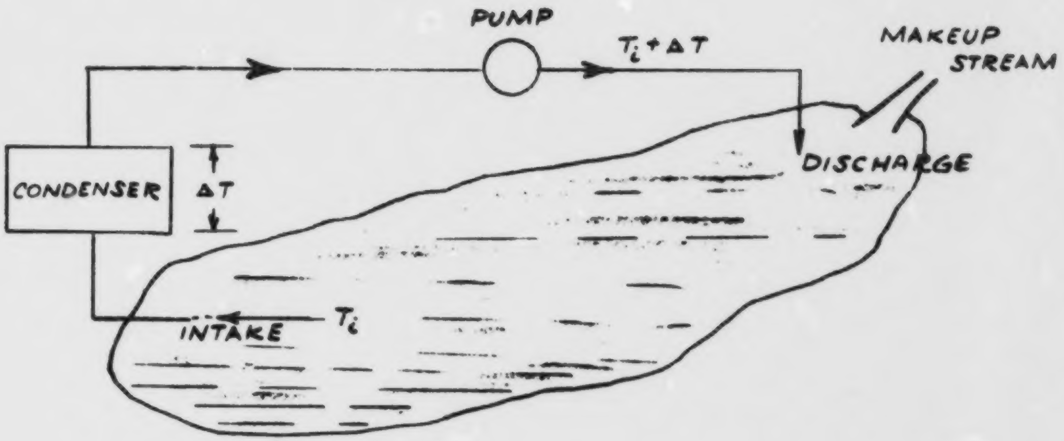


Figure 3.2. Cooling Pond

The heat is dissipated in the same fashion as described in Section 3.2. That is, there is a net positive heat exchange across the water surface effected by conduction, evaporation, and radiation. Cooling capacity must be sufficient to insure satisfactory condenser intake temperatures. Large amounts of surface area are required since the cooling process is so slow. The land cost then becomes the major economic factor in cooling pond cost.

According to Edinger and Geyer (Ref. 3.1) cooling ponds can be classified, on the basis of circulation pattern and temperature distribution, as

1. completely mixed ponds,
2. flow-through ponds, or
3. internally circulating ponds.

The pond circulation patterns are determined by the geometry, placement of intake and discharge, wind patterns, and cooling water flow rate.

3.3.1 Completely Mixed Ponds

A completely mixed pond is one in which the pond temperature is almost uniform except in a small region near the plant discharge. Typically this temperature is just a few degrees (1 - 2°F) above the equilibrium temperature. This situation occurs when the product of the exchange coefficient (K) and the pond area (A) is much greater than the input heat rate (H_P), determined by

$$H_P = \rho C_p Q_P$$

where ρ = water density, (lbm/ft^3)

C_p = specific heat of water, (Btu/lbm-°F)

Q_P = plant discharge rate, (ft^3/day)

Since the temperature (T) is uniform in the pond, the heat balance can be written as

$$\rho C_p Q_P (T_P - T) = KA (T - T_E)$$

The solution is

$$T = \frac{T_P + r\,T_E}{1 + r}$$

where $r = KA/\rho C_p Q_p$

The pond temperature may also be written as

$$\frac{T - T_E}{T_P - T_E} = \frac{1}{1 + r}$$

T approaches T_E as r becomes very large.

The value of the exchange coefficient is usually an average taken over some period of time (day, week, etc.) so that the actual pond temperature will vary with time. A shallow pond will exhibit greater variations in temperature when compared to a deeper pond of equal volume.

3.3.2 Flow-Through Ponds

In a flow-through pond the temperatures decrease away from the plant discharge. Sometimes the pond discharge is returned to a stream or bay to satisfy local temperature regulations. More commonly, the plant intake is located at the low temperature end of the pond.

Since the temperature decreases away from the plant discharge, the heat transfer rate also decreases. Writing a heat balance on a differential element of pond area (dA) yields:

$$-\rho\,C_p Q_P \;\frac{dT}{dA} = K\,(T - T_E)$$

The solution is

$$\frac{T - T_E}{T_P - T_E} = e^{-r}$$

Again,

$$r = \frac{KA}{\rho C_p Q_P}$$

The pond temperature at a point is thus a function of the area between it and the discharge. This analysis is strictly valid for a long narrow pond with no lateral or vertical temperature gradients since a "wide" pond would have vertical and lateral temperature variations.

If we compare the expressions for the temperature excesses of thoroughly mixed and flow through ponds of equal area, we see that

$$\frac{T_m - T_E}{T_f - T_E} = \frac{e^r}{1 + r}$$

where T_m = temperature of mixed pond

T_f = temperature of flow-through pond

The above expression shows that the temperature at the outlet of a flow-through pond is always less than that of a fully-mixed pond of equal area.

3.3.3 Internally Circulating Ponds

An internally circulating pond exhibits a flow pattern in which the warm plant discharge flows on the surface of the pond while the cooled water returns to the plant intake by flowing under the warm layer. Two analyses of such ponds are described in Reference 3.1 with mixing between the two layers included in one and neglected in the other.

3.3.4 Sizing of Cooling Ponds

The cooling ponds must have sufficient surface area that it cools the water to a temperature that will insure satisfactory operation of the power plant. Assuming that we know the flow patterns of the pond and can obtain reliable estimates of the exchange coefficient and equilibrium temperature we can select A such that the pond outlet will provide cool water. Actually it would require a pond of infinite size to cool the warm water to the equilibrium temperature. Usually a 2 - 3° F approach is the lowest practicable in a pond of reasonable size.

A quick estimate of the size of a cooling pond may be made by using the equation

$$H = hA\Delta T$$

H is the heat rejected to the water, ΔT is the temperature difference between the air and average water temperature and a typical value of h is 3.5 Btu/hr-ft^2-°F may be used. McKelvey and Brooke (Ref. 3.6) present a chart with typical values of solar heating and a nomograph to calculate pond size.

"Rule-of-Thumb" sizing is also used with some engineers suggesting 1 acre/Mw + 20% for surrounding land and others suggesting 2 acres/Mw. The former is used for fossil fueled plants while the latter is used for nuclear plants.

Variations in calculated pond size can result depending upon which formula is used. As an illustration, consider the hypothetical 1000 Mw nuclear plant considered in Section 3.2. Plant inlet water temperature is 65°F and discharge is 80°F. For simplicity, K was assumed equal to 150 Btu/ft^2-day-°F, the equilibrium temperature was assumed equal to 63°F and the air temperature to 65°F. The results are given below.

Method of Analysis	Required Pond Size Acres
fully mixed pond	11,700
flow through pond	3,360
$H = 3.5A\Delta T$	5,600
1.2 acre/Mw	1.200
2.0 acre/Mw	2,000

Of course, if the above conditions are changed, the pond will have a new design point and the above comparison may not be valid.

The required surface area of a pond is independent of pond depth for all practical purposes. However, a minimum depth of three feet appears advisable to prevent excessive channeling of flow in ponds with irregular bottoms and to avoid large day-to-day changes in outlet temperature. Some ponds have been designed with deep sections near the discharge and gradual reduction in depth towards the intake.

Conduction to the earth surrounding the pond is almost always neglected but Nelson (Ref. 3.7) has given an average heat transfer coefficient of 0.5 Btu/ft^2-hr-°F. Wet soil such as river bottom mud may be as high as 2.5, however. In any event, it provides a safety factor in the calculations.

Usually natural lakes or bays are not available for use as cooling ponds. The warm discharge water will not be allowed to enter a natural lake at a very high temperature and most natural lakes will not be large enough for the bigger nuclear plants. If a cooling pond is decided upon, an artificial lake must be created by damming and/or excavating. A study must be made to insure that rainfall or runoff is sufficient to provide makeup lost by evaporation or a pumping system must be installed to insure an adequate supply. Maintenance problems are minimal other than checking and up-keep of dams, sluices, and spillways, etc.

Cooling pond costs are a very strong function of the amount of land required. Shade and Smith, using the 1.0 acre/Mw + 20% for surrounding land (at $1000/acre), calculated the cooling pond and piping would cost an additional $5.00/kw. However, they used a more expensive dual pressure condenser in the calculations compared to a simple condenser assumed in the run-of-river calculations. Kolflat (Ref. 3.20) and others (Refs.3.55 and 3.56) have estimated the additional cost of a cooling pond as $2.50/kw. This appears to be a more reasonable figure if we assume 2.0 acre/Mw, $1000 per acre for land and excavation, and $0.5/kw for piping, dams, etc. The cooling pond may prove to be more expensive in the future as larger amounts of more expensive land are required for the big nuclear plants.

3.4 Spray Ponds

Spray ponds operate on the same principle as cooling ponds. However, the evaporation resulting from exposure to the air is enhanced by spraying the warm water into the air over the pond. Residence time in the air is thus increased as is the relative velocity between the water droplets and the air. The spray nozzles are usually located 5 to 10 feet above the surface of the pond. The performance of the pond is a strong function of the design of these nozzles. Poor designs require high pressure and power, do not mix the air and water efficiently (poor spray patterns), and have excessive water losses. Water losses can, however, be cut down with louvered fences. The effect of seven different nozzles on the performance of a spray pond is given in Reference 3.10. Unfortunately, performance is limited by the relatively short contact time between the air and the water spray. Also, impurities can easily enter the open water area and be carried into the condenser, increasing maintenance requirements.

Spray ponds have handled as much as 120,000 gpm. The pumping costs are less than for cooling towers due to the low head requirements. The heads usually range from 4 to 30 feet. A spray pond needs only about 5% of the area of a cooling pond due to the increased transfer coefficient.

There is not a large amount of data on design of spray ponds available in the open literature. Badger and McCabe (Ref. 3.11) found that spray ponds will cool 15 - 20 gallons per hour per square foot when cooling water from 110° F or 120° F down to 70° F. Using the mean values yields a transfer coefficient of 146 Btu/hr-ft^2-°F. Figure 3.3 from Perry (Ref. 3.12) indicates the performance of a spray pond for steam condensing service. Robinson and Roll (Ref. 3.13), using a spray pond 4 feet long and 14 feet long with parallel air and water flow, give the following formula for h in spray ponds:

$$h = 0.0078\, L^{1.5}$$

where L = water load in lb water sprayed/ft^2-hr

Typical values of h range between 20 and 160.

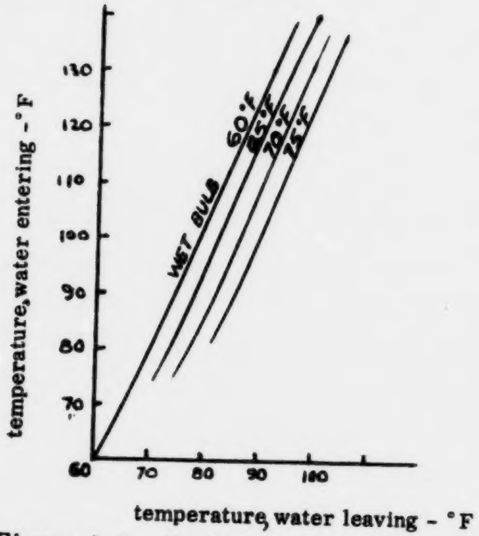


Figure 3.3. Performance of a Spray Pond

Spray ponds require very little maintenance other than occasional cleaning of the pipes and nozzles and routine maintenance on the pump. The danger of freezing and/or poor heat transfer due to climatic conditions may be present. It is most important to start off on the right foot with a proper choice of nozzles and piping layout to obtain efficient spraying. Low heat transfer coefficients, dead zones where the circulation is almost zero, and high power consumption are the undesirable alternatives.

Sizing and capital costs may be estimated as follows. Assuming land at 0.1 acre/Mw and $1000/acre yields $0.1/kw. The spray system (piping, nozzles, pumps and installation) costs about $2.50/kw. Thus, the spray pond would add about $2.50/kw to the base. This price includes the costs of a relatively simple single pass condenser, circulating water pump, screen house, and piping. Other equipment such as dual pressure condensers could raise this figure by several dollars per kw.

3.5 Wet Cooling Towers

3.5.1 Introduction

A wet cooling tower is a device which cools water by bringing the water into contact with air. The water and air flows are directed in such a way as to provide maximum heat transfer. The heat is transferred to the air primarily by evaporation (about 75%) while the remainder is accomplished by sensible heating.

The rate of heat transfer depends upon several factors other than the temperature and enthalpy differences. They are:

a. The interfacial (contact) area between the water and air. Packing is often used to break the water up into smaller droplets which evaporate more easily (splash type packing) and/or provide more surface area upon which to develop liquid films (film type packing).

b. The relative velocity between the air and the water. Higher velocities allow more heat to be carried away in a given amount of time.

c. The residence time of the air in the heat transfer section of the tower. Sufficient packing height should be provided so that the air leaving the tower will be saturated.

A cooling tower is usually comprised of the following parts (Ref. 3.14):

1. Inlet-water distributor
2. Packing
3. Air moving equipment
4. Inlet-air louvres
5. Drift (carryover) eliminators
6. Cooled water storage basin.

In some towers air movement is due to wind or natural draft.

In the past, cooling towers were used primarily as water conservation devices and were required by law in some cases. Presently, as the number of large bodies of water which would serve as heat sinks diminishes and the problems associated with thermal pollution are being realized, cooling towers are being studied as a principal method of satisfactory heat rejection.

3.5.2 Atmospheric Wet Cooling Towers

Atmospheric towers utilize the naturally occurring winds as the air flow. Water is sprayed down into the air stream and the heat is carried away. A sketch of a typical tower is shown in Figure 3.4. Since the air flow is dependent upon natural winds, these towers are usually built with a high, broad side exposed to the prevailing wind. They may be thought of as enclosed spray ponds; the "tower" and louvres keep drift losses down and provide somewhat greater surface area for water films.

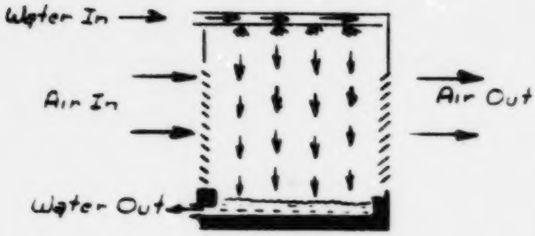


Figure 3.4. Atmospheric Spray-Filled Tower

According to McKelvey and Brooke (Ref. 3.6) they should only be used where the cooling load is less than 1.8×10^6 Btu/hr and the water loading is from 5 to 13 lbs/min-ft^2. These towers are simple and relatively trouble free; packing and filtration equipment maintenance are the most important items. However, they do have several disadvantages:

1. Low or change in direction of winds produce a temperature rise in the exit water.
2. The tower must be installed in an open area where prevailing winds are not obstructed.
3. The cooling range is low.
4. Pumping losses are higher than for atmospheric packed towers which usually do not atomize the water as thoroughly.

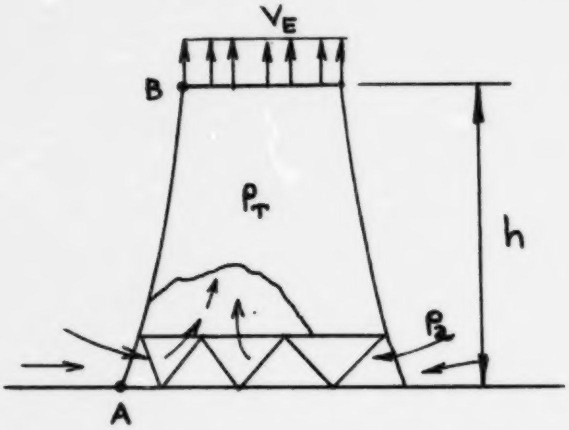
The atmospheric packed tower is another variation of the atmospheric wet tower. These towers are similar to the atmospheric spray tower but are filled with wooden slats in the zone between the spray nozzles and water. As the water drops over the packing, some of it breaks up into fine droplets which fall through the air stream and some forms a film on the packing. This increased contact area increases heat transfer and pumping costs are lower since high pressures are not necessary.

Neither of these types of towers is built today for use in power plants due to the large amount of space (large open surrounding area) required, the relatively high material cost, and the relatively narrow width which necessitates a large support structure to withstand high winds.

3.5.3 Natural Draft (ND) Wet Cooling Towers

The movement of air in these towers is a natural draft due to a difference in density between the inlet and exit air streams. That is, the air outside the tower has a slightly greater (0.005 lb_m/ft^3 is a typical number) density than the air inside the tower above the packing. The difference in hydrostatic pressure between two columns of air of equal height, one inside the tower and the other outside, drives the air through the packing and up the tower. The driving potential is directly proportional to the density difference and the height of the columns.

The concept is illustrated in the sketch below.



ρ_a = ambient air density

ρ_T = air density in tower

V_E = exit velocity

The operation is governed by the following equation

$$\underbrace{(\rho_a - \rho_T)\ \frac{g}{g_o}\ h}_{\text{driving force}} \;=\; \begin{array}{l}\text{pressure drop through packing}\\ +\ \text{frictional losses in tower}\\ +\ \text{kinetic energy of exit air stream.}\end{array}$$

The pressure drop through the packing is due to area contractions and expansions and friction over the packing and spray. It is usually expressed as a resistance in velocity heads (based on inlet air velocity) per foot of packing as a function of water loading ($lb_m/hr\text{-}ft^2$). The resistance increases as the water loading increases and as the inlet velocity level decreases. Obviously, some compromises must be made to obtain sufficient packing to break up the spray or form films for adequate heat transfer without a large pressure drop. The frictional losses in the remainder of the tower are not great since typical towers have about an L/D ratio slightly greater than 1.

The kinetic energy of the exit air stream ($1/2\ \rho_T V_E^2$) is usually specified to insure that the vapor plume will rise away from the tower under all conditions. Typical exit velocities are 10 to 12 fps. In the absence of streamline curvature, the static pressure across the exit plane is equal to the atmospheric pressure at the same height outside the tower.

Therefore, in order to overcome the pressure drops in the tower and provide specified velocities at the exit plane and through the packing (the pressure drop is much greater than the exit kinetic energy loss) in the face of such small density differences, a large height is needed. The latest designs used with large power stations are the huge hyperbolic units with heights of 375 feet being common, both in the United States and England. A sketch of a typical unit is shown in Figure 3.5. The hyperbolic design is due to the fact that as the air enters the base it is moving in the radial direction. As it moves up through the packing, it still possesses some inward radial momentum which causes the air stream to form a vena contracta at some point above the packing. The new hyperbolic design thus closely matches the path of the air flow through the tower while resulting in decreased material costs. In addition, the concrete shell is actually stiffer against wind forces.

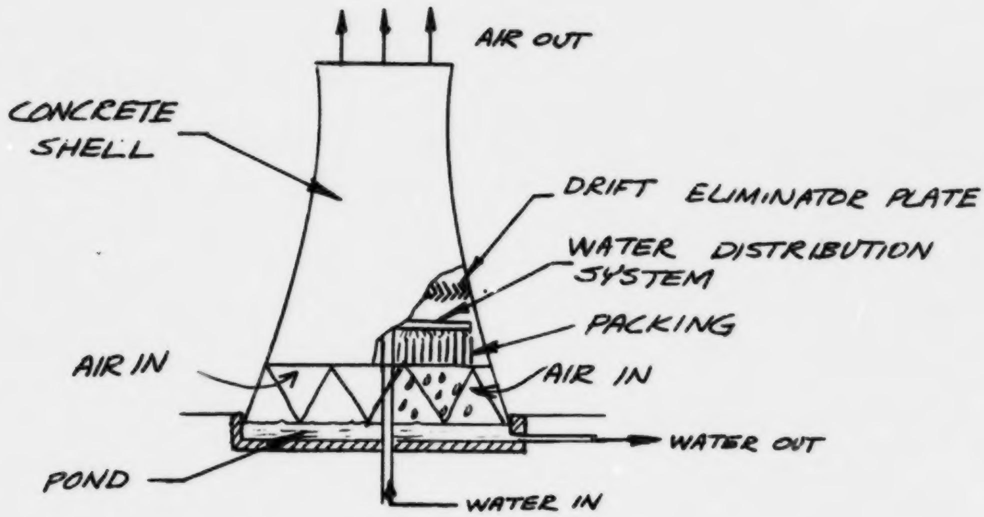


Figure 3.5. Natural Draft-Counterflow Cooling Tower

The latest ND towers designed in this country have ranges which are greater than the approaches by as much as 10° F. Commonly quoted figures (Ref. 3.20) for optimum range and approach for such units are 26° F and 18° F, respectively. The performance is a function of the inlet wet-bulb temperature and relative humidity. The density difference decreases as the inlet wet bulb temperature increases at constant relative humidity while the driving force increases as the relative humidity increases at constant wet bulb temperature. The volume of air that passes through the tower is therefore dependent upon atmospheric conditions. Design point for these towers is nearly always based on a "worst" set of atmospheric conditions. Somewhat oversized towers result but satisfactory operation under adverse conditions is assured.

ND towers may be of the counterflow or crossflow variety. The two types are described in the following sections.

3.5.3.1 Natural Draft - Counterflow

The counterflow tower is depicted in Figure 3.5. The packing is situated so that the air stream flows vertically upwards through the packing while the water is sprayed downward. Either splash or film packing may be employed. The splash packing is usually constructed of redwood timber while the latest film packings (which are more prevalent in the new units) are asbestos sheets separated by plastic spacers. The sheets are generally 3/16" thick on 1-1/2" centers (Ref. 3.24).

The tower is equipped with a drift eliminator plate above the water distribution system to trap water droplets suspended in the air stream. The pressure drop across it is negligible. A typical water distribution system consists of one or two main conduits feeding several thousand nozzles to insure proper distribution of hot water. Pumping head requirements generally run 40 to 50 feet.

3.5.3.2 Natural Draft - Crossflow

The crossflow unit is similar to the counterflow in outside appearance (see Figure 3.6) but the packing is set up around the periphery of the base. About half of the new hyperbolic units are counterflow. They generally use timber splash type packing. When compared to a counterflow unit of given height and diameter, the crossflow unit will typically have a smaller pressure drop and lower cooling capacity due to the lower volume transfer coefficient of the packing.

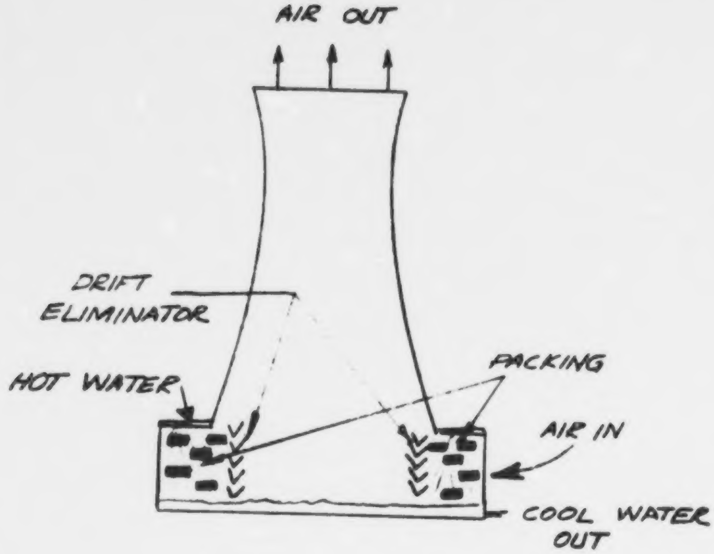


Figure 3.6. Natural Draft-Crossflow Cooling Tower

3.5.4 Mechanical Draft (MD) Wet Cooling Towers

The air flow in a MD tower is produced by a fan which may be placed at either the inlet (forced draft) on the exit (induced draft) of the tower. The air flow rate can thus be controlled to satisfy off-design cooling requirements. The use of the fan also means that the great height of the ND unit is not necessary. Large MD units are 60-75 feet high. Unfortunately, the lower height also means that the problems of ground fogging and/or warm air recirculation may present themselves.

The fan is usually a low-head rise (less than 0.5 inches of water), large volume flow rate unit. Propeller-type fans are by far the most common although some forced draft units do use squirrel-cage fans. Fans on forced draft units are often driven directly or with multiple V-belts while induced draft units use right angle remote drives to remove the motor and gearing from the warm humid exit air stream. The fan does, of course, produce a certain amount of noise, but this has not yet been shown to be a problem.

Drift eliminators are placed at the discharge to prevent most of the suspended water droplets from leaving the tower. Air velocity levels in these towers are in the range 5 to 10 fps. Pumping heads (for the tower alone) for large power station units are 50 - 60 feet.

The initial cost of an MD tower is less than that of comparable capacity ND unit. However, this is offset by higher operating and maintenance costs over the life of the tower.

Mechanical draft units may be classified as:

1. induced draft-counterflow,
2. induced draft crossflow, or
3. forced draft.

They are discussed in the following sections.

3.5.4.1 Mechanical-Induced Draft-Counterflow

A sketch of the configuration and operation of a typical unit is given in Figure 3.7.

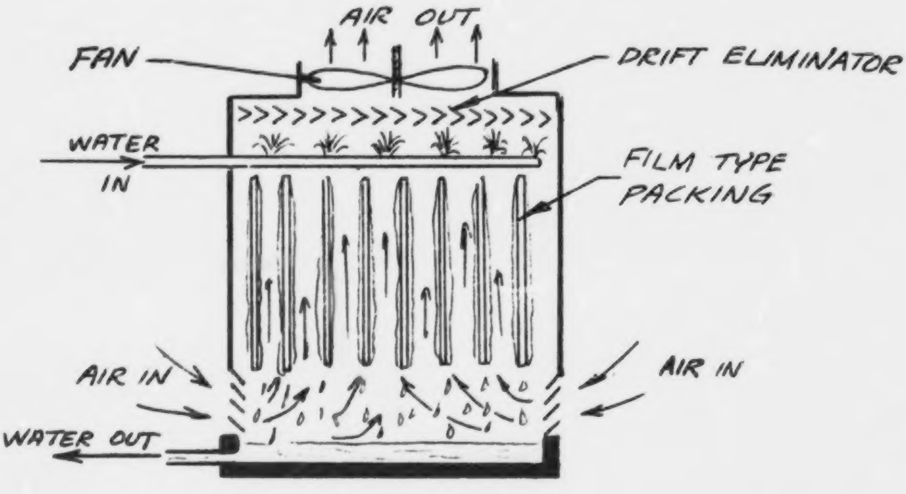


Figure 3.7. Mechanical-Induced Draft Counterflow-Cooling Tower

This arrangement has several advantages:

1. The warmest water contacts the most humid air, while the coldest water contacts the driest air, thus utilizing the evaporative process most effectively.
2. Warm air recirculation is minimized by pushing the discharge straight up.
3. Very large fans (up to 60 feet in diameter) can be used. This avoids going to multiple fans.
4. Close approach and long cooling range is possible (at increased cost).

The disadvantages are:

1. The typically small open area at the base causes high velocity inlet air and therefore increasing the fan horsepower.
2. Baffles must be provided to achieve uniform air velocities through the packing.
3. Requires greater pumping head than a counterflow unit due to taller packing.

The newest counterflow towers use the more efficient film packing similar to that of the ND counterflow towers. The pressure drop through the tower is slightly greater than a comparable crossflow unit since the air must travel upward against the falling water, but the heat transfer per unit volume is greater so that less air is needed. The total horsepower requirements are therefore quite similar.

3.5.4.2 Mechanical-Induced Draft-Crossflow

An induced draft crossflow is illustrated in Figure 3.8.

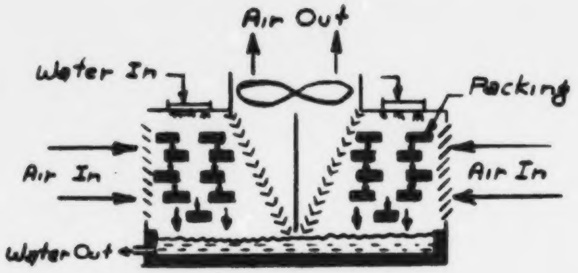


Figure 3.8. Mechanical-Induced Draft-Crossflow Cooling Tower

When compared to a counterflow unit, the crossflow towers have the following advantages:

1. Lower pumping head since the packing is lower.
2. Lower pressure drop through the packing.
3. Higher water loadings are possible for a given height.
4. The overall height of the tower is less.

They do have the following disadvantages, however:

1. A substantial crossflow correction factor must be applied to the driving force, particularly where long ranges and close approaches are desired. In such cases for some pumping heads, a crossflow tower may need more ground area and more fan horsepower than a counterflow tower (Ref. 3.57).
2. The packing is not so efficient and more air flow is required for an equivalent capacity tower.
3. Winter operation is sometimes a problem since ice may form on the inlet louvres and surrounding structure.

4. The packing is in contact with the warmest water causing more rapid deterioration. Unfortunately, inspection is more difficult to perform.

Splash or film type packings are available with these units. More durable materials (asbestos, clay) are available which lengthen the life of packings. The greater initial cost is offset by reduced maintenance costs.

3.5.4.3 Mechanical Forced Draft

The forced draft towers have the fan mounted at the air inlet of the tower. See Figure 3.9.

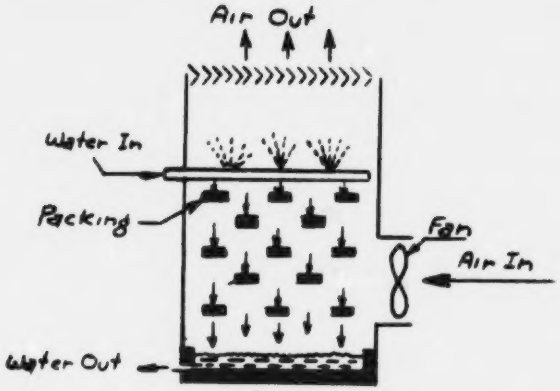


Figure 3.9. Mechanical-Forced Draft Cooling Tower

The advantages of this arrangement are:

1. Vibration and fan noise are reduced since the fan is mounted low near the base.

2. Problems of blade erosion and condensation in gearboxes are greatly reduced.

3. These units are slightly more efficient than the induced draft type since some of the pressure of air velocity is converted into static pressure in the tower and recovered as work.

On the other side of the ledger:

1. Practical fan size is limited (up to 12 feet in diameter) so that multiple fans must be used for high air flow rates.
2. Baffles are necessary to insure uniform air distribution.
3. Recirculation of the hot, humid discharge may be a problem since the air may tend to flow back to the low pressure intake,
4. In cold weather, ice may form on the fan blades, clogging air inlets and sometimes causing breakage.

Forced draft units are not, strictly speaking, either counter- or crossflow but are a combination of the two. In some portions of the tower (usually near the entrance) the air flow is generally perpendicular to the water flow while at points near the exit the flow pattern becomes more similar to a counterflow unit. Splash type packings are usually employed.

3.5.5 Cooling Tower Theory

Cooling tower performance is almost universally based on the theory presented by Merkel (Ref. 3.15) in 1925. He showed how the overall heat transfer (combined sensible and latent) from the water to the air could be calculated using the enthalpy potential as the driving force. The derivation of the basic equations is given first and then modifications for counterflow and crossflow are given.

3.5.5.1 Derivation of Basic Equations

The following derivation is largely taken from Baker and Shryrock (Ref. 3.16) and illustrates the assumptions and methods used in cooling tower analysis. A nomenclature is provided at the end of this section.

Consider an increment of a cooling tower having one square foot of plan area, and a cooling volume, V, with a square foot of exposed water surface per cubic foot of volume. There are L, lbs, of water and G, lbs of dry air per hour flowing through the tower at steady state. A schematic is shown as Figure 3.10.

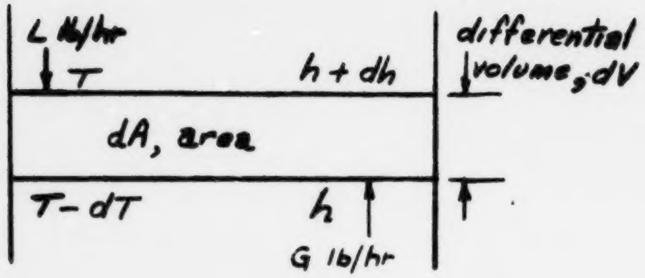


Figure 3.10. Differential Element of Cooling Tower

Within the volume each particle of water is assumed to be surrounded by an interface to which heat is transferred from the water.

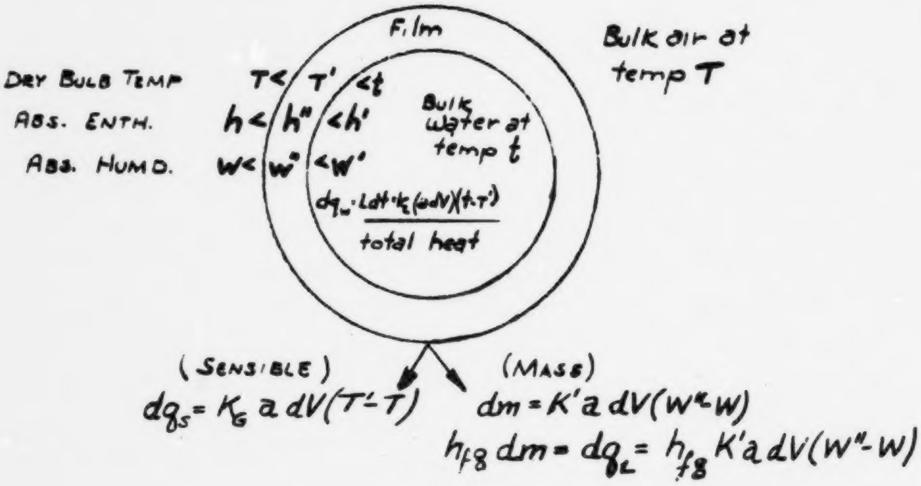


Figure 3.11. Drop of Water Undergoing Simultaneous Heat and Mass Transfer

The rate of heat transfer from the bulk water to the interface is:

$$dq_w = C_{p_w} L\, dt = K_L\, a\, dV\, (t - T') \tag{3-1}$$

A portion of this heat is transferred from the interface to the air as sensible heat. Thus,

$$dq_s = K_G\, a\, dV\, (T' - T) \tag{3-2}$$

The resistance to mass transfer from the water to the interface is neglected, and the mass transfer of vapor from the film to the air is

$$dm = K'\, a\, dV\, (W'' - W) \tag{3-3}$$

This may be converted to a heat rate by multiplying by h_{fg}. Therefore, the latent heat transfer is given by

$$dq_L = h_{fg} K'\, a\, dV\, (W'' - W) \tag{3-4}$$

At steady state, the mass rate of water evaporated equals the rate of humidity increase of the air. Therefore

$$dm = G\, dW \tag{3-5}$$

The heat lost by the water is equal to the heat gained by the air. The heat balance is

$$G (h_2 - h_1) = [L (t_1 - 32) - (L - L_E) (t_2 - 32)]\, C_{p_w} \tag{3-6}$$

$$= [L (t_1 - t_2) + L_E (t_2 - 32)]\, C_{p_w} \tag{3-6}$$

and since $L_E = G (W_2 - W_1)$

$$G (h_2 - h_1) = [L (t_1 - t_2) + G (W_2 - W_1) (t_2 - 32)]\, C_{p_w} \tag{3-7}$$

In practice, the second term on the right-hand side is small and is neglected so that we may write

$$G\,dh = C_{p_w} L\,dt \tag{3-8}$$

The enthalpy of moist air is defined as:

$$h = C_{p_a} (T - T_o) + W \left[h_{fg} + C_{p_v} (T - T_o)\right] \tag{3-9}$$

and

$$dh = (C_{p_a} + W C_{p_v})\, dT + \left[h_{fg} + C_{p_v} (T - T_o)\right] dW \tag{3-10}$$

The specific heat of the moist air is

$$C_p = C_{p_a} + W C_{p_v} \tag{3-11}$$

so

$$dh = C_p\, dT + \left[h_{fg} + C_{p_v} (T - T_o)\right] dW \tag{3-12}$$

and combining Equations (3-8) and (3-12) we can write

$$C_{p_w} L\,dt = G\, C_p\, dT + \left[h_{fg} + C_{p_v} (T - T_o)\right] G\,dW \tag{3-13}$$

The first term on the right represents sensible and the second latent heat. Since

$$dq_s = G C_p\, dT,$$

$$dm = G\,dW$$

and

$$\frac{K_G}{C_p K'} \approx 1$$

for air-water vapor mixtures we can write

$$C_{p_w} L\,dt = K' a\,dV \left\{C_p (T' - T) + \left[h_{fg} + C_{p_v} (T - T_o)\right] (W'' - W)\right\} \tag{3-14}$$

The enthalpy of the air stream may be written as

$$h = C_p T - C_{p_a} T_o + W \, (h_{fg} - C_{p_v} T_o) \tag{3-15}$$

the enthalpy of the interface is

$$h'' = C_p T' - C_{p_a} T_o + W'' (h_{fg} - C_{p_v} T_o) \tag{3-16}$$

Solving Equations (3-15) and (3-16) for T and T', and substituting the results in Equation (3-14)

$$C_{p_w} L dt = K' a dV (h'' - h) + C_{p_v} T \, (W'' - W) \tag{3-17}$$

Merkel showed the second term on the right to be small so it is neglected; thus

$$C_{p_w} L dt = K' a dV (h'' - h) = G dh \tag{3-18}$$

This equation relates the air stream to the interfacial film, the conditions of which are indeterminate for all practical purposes. This is overcome by assuming T' = T. The coefficients K_G and K' are then replaced by overall coefficients K_g and K. Assuming the Lewis relation still applies,

$$\frac{K_g}{K \, C_p} \approx 1..$$

Actually it is more nearly 0.9, but it has been common to assume it does apply. Using these assumptions yields

$$C_{p_w} L dt = K a dV (h' - h) = G dh \tag{3-19}$$

and

$$\frac{KaV}{C_{p_w} L} = \int_{t_1}^{t_2} \frac{dt}{h' - h} \tag{3-20}$$

or

$$\frac{KaV}{G} = \int_{h_1}^{h_2} \frac{dh}{h' - h} \tag{3-21}$$

Equations (3-20) and (3-21) are convertible into one another and are two forms of the basic equation. They conform to the NTU (Number of Transfer Units) heat exchanger design concept in the sense that the value of NTU specifies the size of the equipment necessary to achieve the maximum possible effectiveness. It can be shown that heat exchangers with values of NTU greater than those necessary to produce the maximum effectiveness are too large (excessive amount of heat transfer area) and therefore more expensive than necessary. The NTU corresponding to a set of hypothethical conditions is called the required coefficient. The same calculations applied to a set of test conditions is called the available coefficient of the tower.

The designer must provide sufficient packing depth for good heat transfer effectiveness without excessive pressure drop. The latter point is particularly important in ND towers since the height of the tower increases in direct proportion to the packing pressure drop.

The group KaV/L is often called the "volume transfer coefficient." It is independent of temperature for all practical purposes but is a function of both water and air flow rates. Lowe and Cristie (Ref. 3.58) tested some 60 packings suitable for use in cooling towers and found that the experimental results for all but one could be represented by an equation of the form

$$\frac{Ka}{L} = \lambda \left(\frac{L}{G}\right)^{-n}$$

The two factors λ and n thus characterize the transfer characteristics of the packing. The packing depth (V) is computed to satisfy Equation (3-20).

Nomenclature for Section 3.5.5.1

a = area of water interface, (sq ft/cu ft)

C_p = unit heat capacity (humid heat) of moist air, (Btu/°F-lb dry air)

Cp_a = specific heat of dry air at constant pressure, (Btu/lb-°F)

Cp_v = specific heat of water vapor at constant pressure, (Btu/lb-°F)

Cp_w = specific heat of water at constant pressure, (Btu/lb-°F)

G = air flow rate, (lb dry air/hr-sq ft)

h = enthalpy of moist air, (Btu/lb dry air)

h_1 = enthalpy of moist air, entering tower

h_2 = enthalpy of moist air, leaving tower

h' = enthalpy of moist air, at bulk water temperature

h'' = enthalpy of moist air, at interface temperature

h_{fg} = latent heat of evaporation, assumed constant in system

K = overall unit conductance, mass transfer between saturated air at mass water temperature and main air stream, (lb/hr-sq ft-lb/lb)

K' = unit conductance, mass transfer, interface to main air stream, (lb/hr-sq ft-lb/lb)

K_g = overall unit conductance, sensible heat transfer between main water body and main air stream, (Btu/hr-sq ft-°F)

K_G = overall unit conductance, sensible heat transfer between interface and main air stream, (Btu/hr-sq ft-°F)

K_L = unit conductance, heat transfer, bulk water to interface, (Btu/hr-sq ft-°F)

L = mass water rate, (lb/hr-sq ft)

L_E = mass evaporation loss, (lb/hr-sq ft)

Nomenclature for Section 3.5.5.1 Continued

m = mass transfer rate, interface to air stream,(lb/hr-sq ft)

q_L = rate of latent heat transfer, interface to air stream, (Btu/hr-sq ft)

q_S = rate of sensible heat transfer, interface to air stream, (Btu/hr-sq ft)

q_W = rate of heat transfer, bulk water to interface, (Btu/hr-sq ft)

t = bulk water temperature, (°F)

t_1 = bulk water temperature at inlet (hot water), (°F)

t_2 = bulk water temperature at outlet (cold water), (°F)

T = dry-bulb temperature of air stream, (°F)

T_o = datum temperature for water vapor enthalpy, (°F)

T' = dry-bulb temperature of air at interface, (°F)

T_{WB} = wet-bulb temperature, air stream, (°F)

V = active tower volume, (cu ft/sq ft plan area) (packing height)

W = absolute humidity (humidity ratio) of main mass air, (lb vapor/lb dry air)

W_1 = absolute humidity of main air mass entering tower

W_2 = absolute humidity of main air mass leaving tower

W'' = absolute humidity at interface

W' = absolute humidity, saturated at water temperature

3.5.5.2 Counterflow Tower

The figure below is a schematic of a simplified counterflow tower. The enthalpy of the air increases as it rises in the tower while the enthalpy of the saturated interfacial film on the water drop decreases as the drop falls.

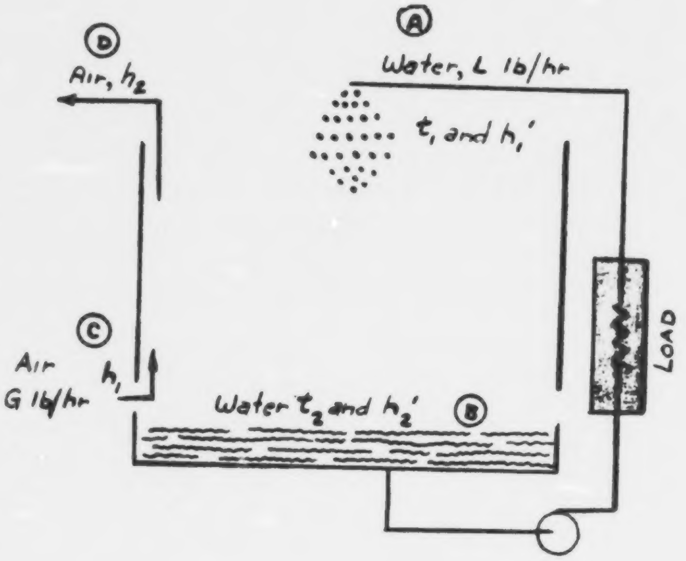


Figure 3.12. Schematic of Counterflow Cooling Tower

A graphic representation of the enthalpies and temperatures for a counterflow unit is given in Figure 3.13. The water entering at the top of the tower (point A) is surrounded by an interfacial film at the bulk water temperature t_1. The enthalpy of the film at point A is h'_1. As the water is cooled, the film follows the saturation curve to point B, where the temperature is t_2 and the enthalpy h_2'.

The air enters with an enthalpy h_1 at point C and leaves with an enthalpy of h_2 at point D. The potential at the base of tower is $h_2' - h_1$, represented by the vertical distance BC. Heat removed from the water is added to the air and from Equation (3-18) $dh = L/G\,dt$. The air operating line is therefore straight.

The heat transfer driving potential at any point in the tower is represented by the vertical distance between AB and CD. Starting at the bottom, since air and water conditions are known there, a numerical integration is performed. The tower is divided into a finite number of volumes with the divisions having equal temperature drops. The equation

$$\Delta q = C_p L \Delta t = KadV \, (h' - h)_m$$

applies to the increment where $(h' - h)_m$ is the mean enthalpy difference in the increment of volume and delta refers to quantities for the increment. Summation of the increments yields

$$KaV = L C_p \Delta t \sum \frac{1}{(h' - h)_m}$$

Illustrations of the use of this method are given in References (3.16) and (3.17).

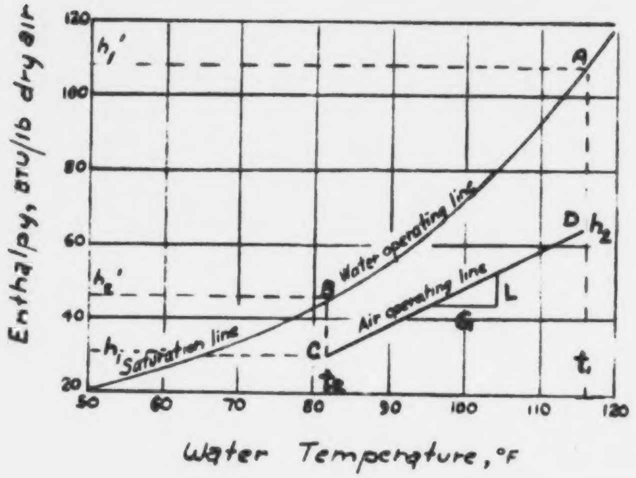


Figure 3.13. Enthalpy-Temperature Diagram of Air and Water in Cooling Tower

3.5.5.3 Crossflow Tower

A crossflow tower has horizontal and vertical variations in potential. The cross sections are divided into unit volumes Δx wide and Δy high. The mean driving force in crossflow cooling is calculated by averaging the reciprocals of the inlet and exit potential differences. This straight line relationship is a necessary assumption because calculation of the mean driving force is not easily done since each unit-volume is as complex as the tower as a whole. Excellent discussions and examples are given in References 3.15 and 3.57.

3.5.5.4 Discussion and Summary

The analysis presented here included several assumptions:

1. The bulk water and interface were assumed to be at the same temperature.
2. The resistance to mass transfer from the water to the interface was negligible.
3. The quantity of water evaporated during the cooling process was neglected.

The first two assumptions indicate that the calculated energy transfer will be higher than can be verified by experiments. The effects become more pronounced at greater temperature differentials between the air and the water and at higher L/G ratios since evaporation effects become more important at the higher L/G ratios. This is the reason for the use of "hot water correction factors," discussed in References 3.18 and 3.19.

3.5.6 Size Estimation

The size of a wet cooling tower is dependent upon four factors: (1) heat load, (2) range, (3) approach, and (4) wet bulb temperature. The heat load (in Btu/hr) can be calculated from the following equation

$$H = 3413 \times \left[\frac{\eta_p}{\eta_{th}} - 1.0 \right] \times \text{(plant size - kw)}$$

where η_p = plant efficiency (0.95 for nuclear; 0.85 for fossil fuels)

η_{th} = thermodynamic efficiency of the Rankine cycle.

The range is usually specified by the increase in temperature through the condenser or local temperature regulations. The approach is generally governed by economics --the smaller the approach, the greater the capital cost of the unit. Usually a 5° approach is the coldest water outlet temperature that a cooling tower manufacturer will guarantee. Even this figure is seen only in very well designed mechanical draft towers. Natural draft units run with 15 - 20°F approaches. In sizing a tower, approach, followed closely by gpm is the most important factor. The wet-bulb temperature usually used for design purposes is one somewhere near the maximum for year-round variations. It can be higher than the value obtained from meteorological data if other towers or process equipment are in the immediate vicinity.

An important parameter is the circulation rate (L) which is governed by the heat load and range according to the following equation

$$L \text{ (gpm)} = \frac{Q \text{ (Btu/hr)}}{\text{range} \times 500} .$$

The circulation rate is therefore inversely proportional to the range. Addition of water is required to replace water loss by evaporation, carryover, blowdown, and leakage. This makeup amounts to about 3.5% of the circulation rate with about 1.5% due to evaporation and the majority of the remainder attributable to blowdown. Properly designed drift plates practically eliminate carryover.

Natural draft towers are necessarily larger than mechanical draft types. A rough idea of the relative plan areas may be made by comparing packing water loads. Natural draft units run with 2 to 4 gpm/ft^2 while mechanical draft towers average about 6 gpm/ft^2. It is possible to achieve up to 10 gpm/ft^2 with a doubleflow MD tower. A natural draft tower will therefore require 2 to 3 times more ground area than an equivalent capacity MD unit.

There is an even greater contrast in the heights of the two towers. Large counterflow MD towers require about 20 feet below the packing for inlet area and 5 to 10 feet above the packing for water distribution system, fans and stacks. The packing for a large power station tower will be 20 to 30 feet high. Using the maximum values gives 60 feet high.

As shown in Section 3.5.3, the great height of the ND tower is necessary to overcome the losses thru the packing, in the tower and at the discharge. Assuming an average density difference of 0.005 lb_m/ft^2 and a total pressure loss of 0.4 inches of water (including the kinetic energy loss at the exit) yields 416 feet for the required height. The density difference could represent an ambient condition of 87° F with relative humidity of 50% and a discharge at 100° F, 100% relative humidity. This discharge temperature may sound high, but the large hyperbolic tower at Ft. Martin, West Virginia cools water from 114° F to 90° F, for example. The figure of 0.4 inches of water includes a packing pressure drop of about 3.7 velocity heads (based on 3 fps) per foot of packing for a 40 foot deep packing based on data obtained from the same tower (Ref. 3.59).

Tower plan area estimates were made for a 1000 Mw nuclear plant using 0.8 gpm/kw. The following required base areas were calculated:

Type	Loading	Required Area	Area/2
ND	3 gpm/ft^2	2.67×10^5 ft^2	1.33×10^5
MD	6 gpm/ft^2	1.33×10^5 ft^2	0.67×10^5

The required area of the ND tower could be provided by two units 412 feet in diameter. The MD required area could be provided by a combination of 10 square cells. The dimensions would then be 115 feet wide and 1150 feet long.

A size comparison has been given by Kolflat (Ref. 3.20). A hyperbolic natural draft tower for a 500 Mw station would be about 370 feet high and have a 400 foot base while a mechanical draft unit for the same station would be about 600 feet long, 70 feet wide and 60 feet high. This yields the following base areas:

ND	1.26×10^5 ft^2
MD	0.42×10^5 ft^2

These figures compare favorably with the example given above (compare with Area/2 Column).

In addition to the area required for the tower itself, land is required for reservoirs, blowdown ponds, and pumping and storage areas. Large towers cannot be packed too closely together since high winds produced by a Venturi-effect can cause excessively high wind loads.

The size of a cooling tower can also vary with location and design capability. As the meteorological conditions vary from place to place, tower performance (especially wet natural draft units) will be altered. The size can also be reduced if a risk is taken that the cooling tower will not be able to accommodate the design heat load 100% of the time. A reduction in capacity is then incurred under adverse conditions. Reference 3.21 describes a method of approaching this problem and calculating the possible reduction in capacity.

3.5.7 Cost Information

Cooling tower costs can be grouped under one of three headings:

1. Initial Costs,
2. Annual Fixed Costs, and
3. Annual Operating Costs.

The initial costs are a strong function of the plant site. Land costs, installation and accessory costs, and design ambient conditions may all be affected by location. In some locations it may be necessary to construct reservoirs and dams to insure sufficient makeup water supply while in other locales rivers or lakes may be available. Since a cooling tower installation with, for example, run-of-river makeup will not require the more expensive intake and discharge tunnels, screen house, screens and rakes that a run-of-river once-through cooling system requires, the cost data will be presented for the complete cooling system of condenser, tower, piping, etc. From this point of view the cooling system may be considered as a piece of equipment which will be interfaced with the plant at the turbine exhaust.

There are two types of cost information available in the open literature. One gives ball-park estimates usually expressed as \$/kw while the other gives the cost in dollars expressed as an equation or graph. Both of these types of numbers are derived from correlations of past data, but the latter type (typically from process mechanical draft units) is expressed as a function of the heat load, range, approach, and wet-bulb temperature so that the effect of changing conditions may be studied.

The following table gives a breakdown of cooling tower costs.

Table 1

COOLING TOWER COSTS

1. Initial Costs
   - a. Capital Cost
   - b. Installation Cost
   - c. Accessory Capital Costs
2. Annual Fixed Costs
   - a. Amortization and Depreciation
   - b. Interest Costs
   - c. Taxes
   - d. Insurance
   - e. Rents
3. Annual Operating Costs
   - a. Power (Pump and Tower Fan)
   - b. Power (Other)
   - c. Maintenance
   - d. Water Treatment
   - e. Makeup Water
   - f. Sewer Charges

The approximate initial costs for the ND and MD towers are discussed in the following sections. In Task I, Phase II, we shall carefully correlate the most recent data on cooling towers and determine cost-size relationships for use in the system studies.

The bulk of annual fixed costs is also a strong function of the particular plant. Taxes, insurance, rents can vary widely from area to area. Interest charges are usually the same across the country but may vary from year to year. Capital costs for a power plant are generally amortized over 30 years.

Operating and maintenance costs for pumps vary as to location and type of tower. According to Kovats (Ref. 3. 3), operating costs for different brands may be taken as identical and does not depend on whether large single or smaller multiple setups are used. Downtime is greater with large single, however, and four pumps in steps of about 32, 67, 82 and 100% of total flow gives good range of flow adjustment and flexibility.

Kovats also says that pumps used in cooling tower service require 1.0 to 1.2 percent of generator load capacity and Berg and Larson (Ref. 3.26) suggest that maintenance costs can be taken as equal to power costs for a first estimate. Accurate operating and maintenance data is not available from manufacturers due to lack of feedback from users. Over the 30 year life of a pump, approximately another 2/3 of the initial cost will be spent on replacement parts.

Costs of water treatment are given by Lane and Larson (Ref. 3.27), Jones (Ref. 3.28), and Berg and Larson (Ref. 3.26). Lane and Larson present costs for lime softening, zeolite softening. Jones draws attention to the much lower costs attainable when the unit is run at higher concentrations. Discussion of the limits of amounts of certain chemicals, particularly calcium and sulfates, is also given by Lane and Larson (Ref. 3.26).

3.5.7.1 ND Towers - Capital Cost

Some recent economic studies have indicated that the wet ND hyperbolic design is the most economical tower when such a unit is required. Unfortunately, only a few of these have been built in the United States and fifteen or so are in the construction stage so a large amount of data is not available. The British have built some 60 to 70 of these towers but labor and material costs are quite different and design wet-bulb temperatures run 15 to 20 degrees lower than in the United States. Due to the higher wet-bulb temperatures, U. S. plants with hyperbolic towers usually run at higher exhaust pressures (2. 5 - 3. 5 inches Hg) and resultant lower efficiency. This is necessary to achieve suitable ranges and practical approaches.

It must be noted that the studies referred to above are overall studies which included fuel, operating, and usually transmission costs. A typical situation occurs where the use of a cooling tower allows the plant to be located on less expensive land or closer to a mine mouth or a large user of electricity. Water around mine mouths is generally acidic and requires treatment before use in the condenser. A cooling tower that recirculates the water allows treatment to be performed economically. That is, many of the studies show that cooling towers are attractive due to reduced fuel, land, or power transmission costs.

The following table is a compilation of cooling system costs using ND wet cooling towers.

Table 2

Ref.	Plant Size	Fuel	$/kw	Comments
3.20	1000 Mw	N	15.00*	Hypothetical plants
3.23	1000 Mw	N	11.50*	
3.24	540 Mw	f	8.65*	Ft. Martin plant
3.2	1800 Mw	f	7.50	Run-of-river makeup
3.60	1800 Mw	f	7.90*	Keystone plant
3.61	1600 Mw	f	9.40	Reservoir makeup

---

*$4/kw was added to tower cost for condenser, piping, pumps, etc. to compute system cost.

A study on cooling towers for a 400 Mw fossil-fueled plant cooling 150,000 gpm (Ref.   ) yielded the following costs as a function of approach.

Tower Diameter	Approach, °F	Tower Cost, $/kw	System Cost, $/kw
210	21.6	3.0	7.0
230	18.8	3.43	7.43
250	16.5	4.12	8.12
270	13.9	4.70	8.70

If we assume a 38% thermal efficiency, the range is 22.6°F. The above table clearly shows the increasing cost of closer approaches. The study does not account for any increases in cost due to plant location.

The estimated cooling system cost using ND wet cooling towers is summarized below.

Fuel	Approach, °F	System Cost, $/kw	Makeup
f	22	7.00	river
f	18	7.50	river
f	18	10.50	reservoir
N	18	11.50	river
N	18	15.00	reservoir

The cost of pumping runs about 0.4% of generator capacity for a nuclear plant. The tower has an 18°F approach and a 25°F range. The cost under other conditions may be predicted from graphs given in Section 3.5.7.2.

3.5.7.2 MD Towers - Capital Cost

Prior to 1962, MD draft towers dominated the power plant cooling tower field. Recently, however, the large power plants have been equipped with ND hyperbolic towers. In spite of their higher initial cost, ND towers have lower operating and maintenance costs. Pumping costs will be similar for both towers. Fans for MD units consume about 0.8% of generator capacity (Ref. 3.20). This may be even higher if a very close approach (less than 10°F) is desired. ND towers do not require any fan power. The use of a mechanical draft tower results in a penalty of 2 to 3% of plant capability compared to 1 to 1.5% for a natural draft unit.

Kolflat (Ref. 3.20) has estimated that an induced draft wet MD tower for a nuclear plant would add $6/kw to the plant cost. Cooling towers for the 540 Mw nuclear plant at Vernon, Vermont would cost an additional 6 million dollars or $11.1/kw (Ref. 3.22). The Vernon towers have a very close approach which accounts for the high cost. The effect of approach on initial cost may be seen in the following data taken from Ravet (Ref. 3.62):

400 Mw fossil fueled plant 150,000 gpm $T_{WB} = 76$ 53% R. H.

Approach, °F	Tower Cost, $	$/kw
21.4	4.8×10^5	1.20
16.6	6.0	1.50
13.6	7.2	1.80
12.0	8.0	2.00
9.6	9.6	2.40

Lockart et al (Ref. 3.63) correlated the data obtained by Knowlton (Ref.3.64) for 52 mechanical draft cooling towers. The data are presented in the form of a graph on Figure 3.14. It was necessary to adjust the costs since the data was obtained in 1955 and some preliminary calculations indicated that costs have approximately doubled since then. Extrapolation of the curves was also performed to extend the useful range.

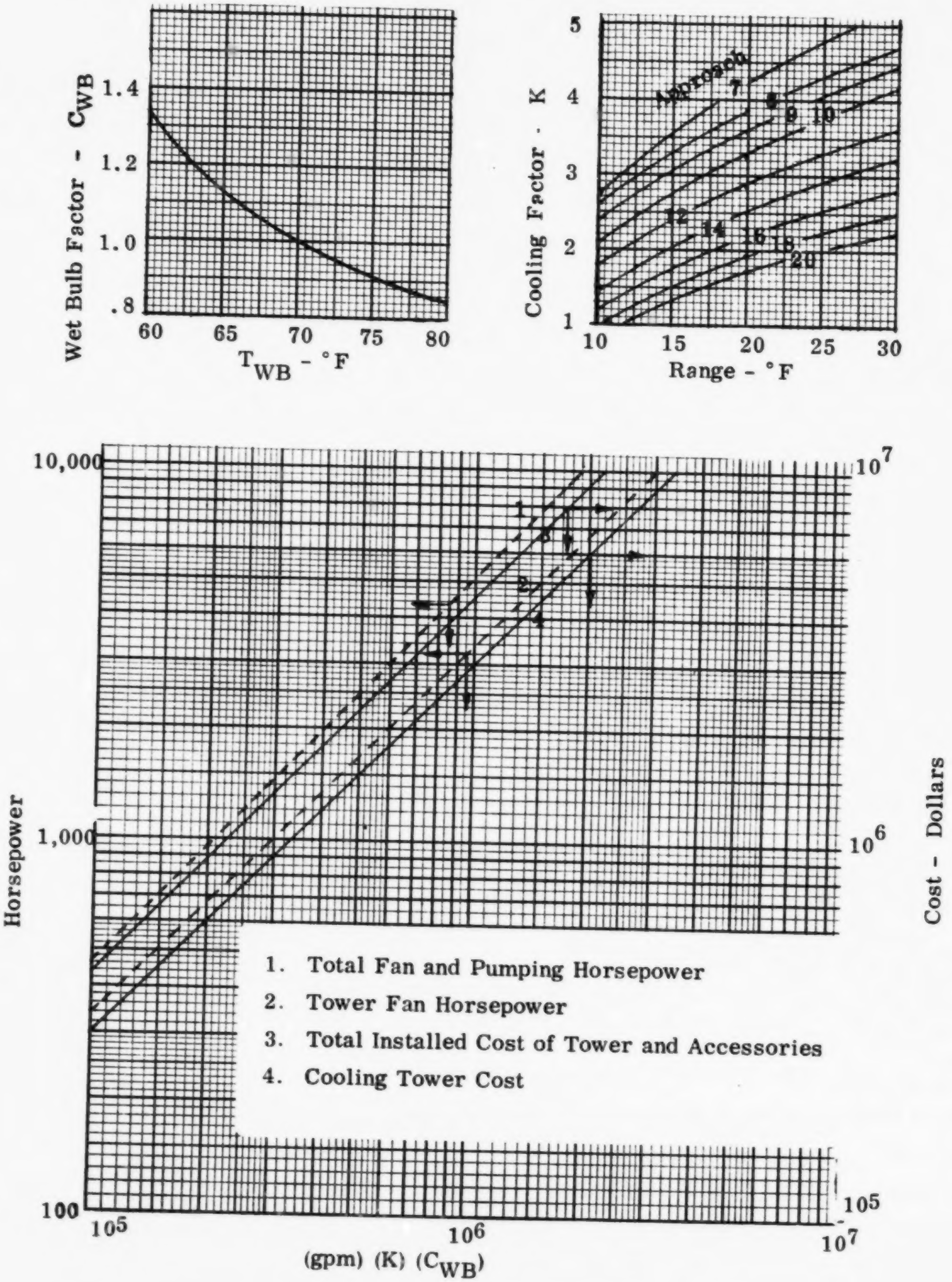


Figure 3.14. Costs and Power Consumption of MD Wet Cooling Towers

To illustrate use of the curves, we have compared data from the graphs with Ravet's data.

$T_{WB} = 76 \quad \therefore \quad C_{WB} = 0.90$

Range $= 23°F \quad$ gpm $= 1.5 \times 10^5$

Approach, °F	K	gpm x K x C_{WB}	Tower Cost, $	$/kw	Tower Cost, (Ref. 3.62)
21.4	1.8	2.43×10^5	7.2×10^5	1.80	4.8×10^5
16.6	2.35	3.17×10^5	9.5×10^5	2.37	6.0×10^5
13.6	2.90	3.92×10^5	11.8×10^5	2.94	7.2×10^5
12.0	3.13	4.23×10^5	12.5×10^5	3.12	8.0×10^5
9.6	3.70	4.99×10^5	15.0×10^5	3.74	9.6×10^5

The costs obtained from the graph appear to be about 50% high.

If the 400 Mw plant was a nuclear installation, the gpm would be approximately doubled. The graphs yielded the following figures:

Approach, °F	Tower Cost, $/kw	Total Cost, Installed Tower & Accessories, $/kw	Total System Costs, $/kw
21.4	3.60	5.37	9.37
16.6	4.75	7.13	11.13
13.6	5.87	8.88	12.88
12.0	6.30	9.55	13.55
9.6	7.50	11.25	15.25

The cost figures given by Kolflat and that of the Vernon, Vermont plant lie within the range given in the third column. It is felt, then, that Figure 3.14 will deliver reasonable estimates of wet mechanical draft cooling tower initial costs. The total system cost at 18°F approach is found to be $10.50/kw. This compares to $11.50/kw for an ND unit with run-of-river makeup or $15.00/kw for ND tower with reservoir makeup (figures are for nuclear plants).

Kolflat has also estimated that the fans for a wet mechanical draft unit will consume about 0.8% of the generator capacity. Figure 3.14 gives 0.46% of generator capacity for the fan alone and 0.85% for the fan and pump horsepower requirements. This figure is for a nuclear plant with a design wet-bulb temperature equal to 72° F, 25° F range, 18° F approach and 0.85 gpm/kw. A fossil-fueled plant operating under the same conditions would use about 0.23% for fans alone and 0.42% for the total of fans and pumps.

The power costs are a strong function of approach. If the approach was 10° F instead of 18° F for the nuclear plant, the consumptions would be 0.78% and 1.41% for the fans and the fans and pumps, respectively.

The pumping cost for a natural draft tower may be estimated from Figure 3.14 by subtracting the fan horsepower requirement from the total. This is valid since the pumping heads are quite similar for both types of towers.

3.5.8 Maintenance Considerations

It has been pointed out by a number of authors that proper maintenance is essential to the satisfactory performance of a cooling tower. The task is difficult. Components of the tower may be flooded, splashed, sprayed, exposed to hot humid air and yearly variations in ambient conditions. The capacity for heat transfer is dependent upon water flow, air flow and the particular tower characteristics. Poor maintenance can produce adverse effects in all three areas.

Water flow is often altered, particularly the distribution system. Flooding due to corrosion or poor spray patterns due to corrosion or clogging are often seen. Effects on the air flow are usually exhibited by increased static pressure. Scale, oil, mud, and algae accumulated on the fill or mist eliminators result in decreased air flow and poor performance. Flow pulsations within the fan sometimes occur causing unstable performance. Air leaks may decrease the capacity of the tower by allowing air to flow through the tower without contacting the falling water. Within the water, deterioration of the packing and support structures can cause unfavorable distributions of air and water, decreasing the heat transfer coefficients. A long list of examples of poor maintenance is given in Reference 3.29.

The key to success in this area is periodic maintenance and inspection. Present day corrosion inhibitors and fungicides have extended the time required between replacement of items such as packing, fastners, and support structure. A periodic maintenance schedule allows work to be accomplished under the most favorable conditions.

Proper selection of materials is very important as are the particular coatings and treatments selected. A compromise between long service life and cost must be made. A set of "normal" operating conditions has been defined to determine service life for materials and costs (Ref. 3.30).

1. Circulating water pH between 6 and 8; chloride content below 750 ppm as NaCl; sodium bicarbonate content below 200 ppm as $NaHCO_3$; maximum temperature of 140F; no significant contamination with unusual chemicals or foreign substances; and adequate water treatment to minimize corrosion and scaling.

2. Chlorine, if used, is added intermittently with a free residual of one ppm or less maintained for short periods.

3. Atmosphere around tower is not worse than moderate industrial. Rainfall and fog are only slightly acid and do not contain significant chlorides or hydrogen sulfide.

   With regard to chlorine treatment, the following should be noted: "The surface deterioration of redwood in cooling tower service may be virtually eliminated by avoiding the use of chlorine and using a non-oxidizing algaecide in the treatment of the circulating water."

Reference 3.31 lists the standards for lumber, plastics, electrical service, metal hardware, and safety usually considered in specifications of cooling towers. The costs of maintenance have been discussed in Section 3.5.7.

3.5.9 Secondary Pollution Considerations

The use of a cooling tower may be accompanied by undesirable side effects. Chemicals harmful to various forms of life may be discharged to nearby streams, rivers, or lakes. The evaporation of water into the air stream results in an increase in the concentration of dissolved solids in the remaining water within the tower. Scale formation or corrosion in various components of the cooling system will eventually occur depending upon the pH of the water. Scale formation generally lowers the heat transfer ability of the system while corrosion leads to material failures. Briefly, the saturation pH (pH_s) is the minimum pH that a sample of water can have at a certain temperature and still precipitate scale. If the pH of the water is above pH_s, scale will be produced and the water is essentially non-corrosive. However, if the pH is below pH_s, scale (if any) will be removed and the water may be corrosive. The lower the actual pH relative to pH_s, the greater the tendency of the water to be corrosive. To maintain the pH of the recirculated water between 6 and 8, sulfuric acid or caustic soda is commonly added to the makeup water to prevent scaling or excessive corrosion. Scaling, corrosion, and wood deterioration all increase with increasing temperature.

To maintain the proper pH and keep the concentrations of dissolved solids below certain levels, a portion of the recirculated water is bled-off and discharged as waste. It is replaced with raw or treated water. This process is known as blowdown and may be performed continuously or intermittently.

A typical water treatment program is made up of the following five steps:

1. Analysis of the available raw water.

2. Determination of maximum allowable concentration of dissolved solids.

3. Determination of the optimum concentration for the system. This involves a tradeoff between the additional cost of water treatment and cost of deterioration of equipment due to scale and corrosion.

4. Determination of the optimum pH of the water system.
5. Selection of the proper scale and corrosion inhibitors. It is possible to select an inhibition that will reduce corrosion in one part of the system and also prevent scale deposition in the hotter portions of the system.

Malley and Gasper (Ref. 3.65) describe sixteen systems for bleeding cooling tower water basins. They are divided into three categories:

1. Continuous bleed-off,
2. Proportional or semi-proportional bleed-off, and
3. Automatic units.

The first two systems utilize manual checking and adjustment of the concentration level while the third monitors concentration via a conductivity measurement and regulates bleed-off through a feedback control system.

The continuous systems generally have a pipe, trough, or weir somewhere in the system. A portion of the water in the basin is allowed to flow out of the basin and is discharged as waste. Some systems collect a portion of the water that has just fallen through the packing and discharge it to the waste stream. Bleed-off may also be taken from the circulating pump. The proportional systems permit a bleed-off which is proportional to cooling load. They usually have valves whose activity is tied to circulating water pump or fan operation.

According to Berg, et al (Ref. 3.26) the blowdown in gallons per hour required to satisfy a given concentration ratio may be estimated as a fraction of the evaporative loss from the following equation

$$\text{Blowdown} = \frac{0.06 \times R \times L}{C - 1}$$

where R = range, (°F)
L = circulation rate, (gpm)
C = ratio of circulation water to raw-water minerals concentration.

Values of C run from 2 to 10. For example, a large ND cooling tower might have the following values:

$$R = 24^\circ F$$

$$L = 250,000 \text{ gpm}$$

$$C = 5$$

$$\therefore \text{ Blowdown } = \frac{0.06\,(24)\,(250,000)}{4} = 90,000 \text{ gal/hr}$$

This water requirement must, of course, be added to the amount of water loss via evaporation and other operations.

Careless handling of blowdown and corrosion inhibitors can discharge pollutants into the stream. This discharged water may contain high concentrations of acids, caustic, or chlorine (used for control of algae and slime growth). Chlorinated phenols and quaternary salts are also used for algae and slime control. Chlorine is not usually a problem since halogens are oxidizing agents and dissipate quickly. The salts are toxic but can be reduced to low concentrations and biodegraded quite easily. The chlorinated phenols, on the other hand, are not as easily degraded and remain toxic for long periods of time.

In cases where very stubborn growths are observed heavy metal salts such as copper or mercury compounds may be used. These materials remain toxic as long as they are water soluble. This causes organisms to demand even more oxygen from the already oxygen deficient water. Higher temperatures also tend to raise the toxicity of many materials.

Abnormal situations can enter the secondary pollution picture. During startup, lignin, chromated copper arsenate (the most popular preservative), and other preservatives may be leached out of wood very rapidly and enter the waste stream in high concentrations. Use of asbestos and plastics which offer better resistance to fungal attack alleviate this problem to some extent. Heavy-duty coatings such as epoxy coal tar may have been used on steel and cast iron components. Polychlorophenates have been used for years as an algaecide and fungicide in concentrations up to 300 ppm as a shock treatment (Ref. 3.66). Dust and other materials from the air stream can collect in the basin of the cooling tower. If not removed, they may enter the stream. Acid cleaning of cooling equipment is also a potential problem unless the acid is properly drained and diluted before disposal.

Cooling towers can be set up so that the rate of pollutants entering the streams is within some accommodation limit. Operator and equipment failures can destroy this balance by releasing large amounts of pollutants in a short amount of time. Several methods of sampling the waste stream involving chemical or electrical measurements are available. The sampling and analysis must be done rapidly so that adequate corrections may be made. The control methods are dependent on the comparative sizes of pollutant discharge and receiving stream. If the amount of pollutants is small, dilution combined with controlled release rates can be used. If the ratio is large, coagulation, neutralization, or softening must be employed. Under severe conditions it may be necessary to shut down the tower for several hours while the water is treated.

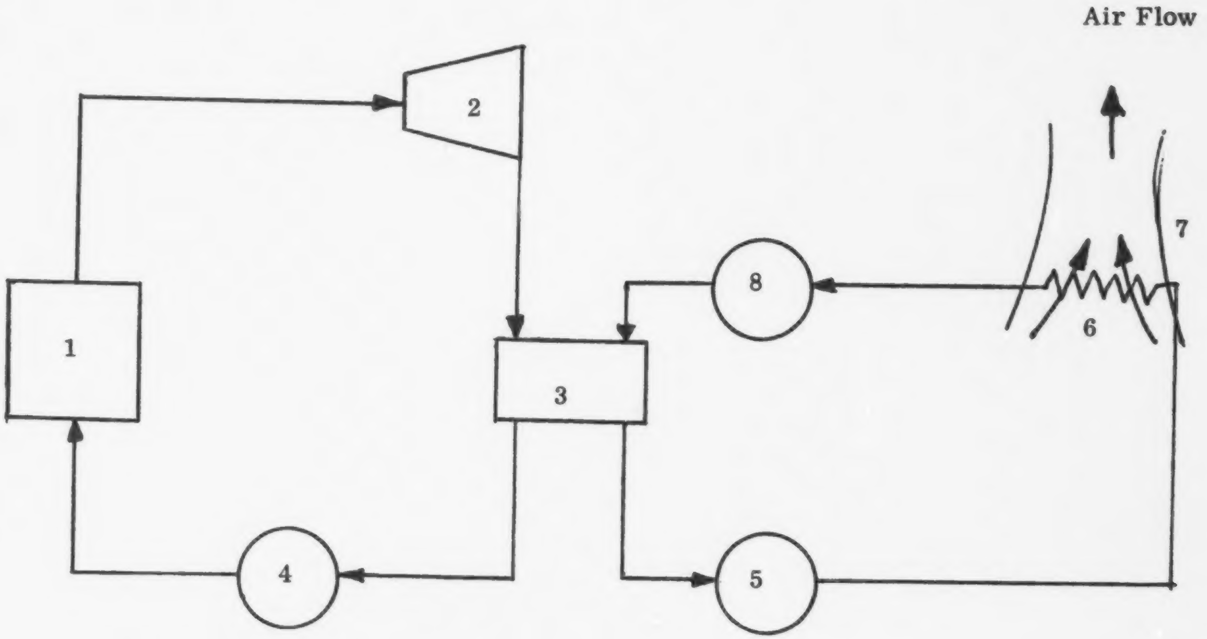
3.6 Dry Cooling Towers

3.6.1 Introduction

The dry cooling tower is a relatively recent development. Only a few power plants have been built which utilize this cooling technique. Notable examples are a 120 Mw plant at Rugeley, England and a 15 Mw plant in West Pakistan. The dry cooling tower technique appears to be a very promising alternative to once-through cooling.

3.6.2 Description

The dry cooling tower is usually employed as a component of the "Heller System" which is illustrated in Figure 3.15. The turbine exhaust is condensed in a direct contact spray condenser. A portion of the condensate flows to the boiler, while the remainder is pumped through the dry cooling tower in a closed loop, returning to the condenser. The dry cooling tower itself consists of an air-cooled heat exchanger mounted inside a cooling tower chimney. The airflow past the heat exchanger may be driven by fans (forced draft), or it may result from the natural draft of the heated air inside the tower. The natural draft tower requires a larger tower than the forced draft, but it requires no mechanical air moving equipment.



1. Boiler
2. Turbine
3. Direct Contact Spray Condenser
4. Feedwater Pump
5. Cooling Loop Water Pump
6. Air Cooled Heat Exchanger
7. Cooling Tower Chimney
8. Water Turbine

Figure 3.15. Basic Diagram of the "Heller System," A Direct Contact Spray Condenser With a Dry Cooling Tower

The heat flow from the water to the air is dependent only upon the mechanisms of conduction and convection; no evaporation takes place within the tower. The heat exchanger is generally a fin-tube, cross-flow type.

The direct contact spray condenser consists of a steel shell into which the turbine steam is exhausted. Cool water from the cooling tower is introduced into the condenser as a spray of water droplets from a series of evenly distributed nozzles. Upon contact with the water spray, the steam condenses and is cooled to a temperature nearly equal to the temperature of the spray water. The condensate and cooling water collect at the bottom of the condenser. This water is then pumped to the boiler and through the cooling tower as indicated in Figure 3.15. This type of condenser is used because it has a much higher thermal efficiency and can be made much more compact than conventional condensers. However, since the cooling water mixes with the condensate, the cooling water must be satisfactory for use in the boiler.

The pressure in the condenser is always well below atmospheric. In order to prevent air leakage into the cooling water through joints in the piping and heat exchanger, the water is pressurized to a level higher than atmospheric by a pump. This represents an additional auxiliary power cost for the systems. However, in large systems a hydraulic turbine can be placed before the condenser to recover some of the energy required for pressurization.

3.6.3 Theory of Operation

The heat transfer between the water and the air in the heat exchanger portion of the system, when a cross flow heat exchanger is used, is governed by the following equation. (See Figure 3.16.)

$$Q = UAF_G \left\{ \frac{(T_1 - T_3)-(T_2 - T_4)}{\ell n \left[\frac{T_1 - T_3}{T_2 - T_4}\right]} \right\} \qquad (3\text{-}22)$$

where Q = heat transfer rate, (Btu/hr)

A = heat exchanger surface area,(ft^2)

F_G = cross-flow correction factor, (≈ 1.0)

T_1 = inlet water temperature, (° F)

T_2 = outlet water temperature, (° F)

T_3 = inlet air temperature, (° F)

T_4 = outlet air temperature, (° F)

U = overall coefficient of heat transfer, (Btu/hr-ft^2-° F)

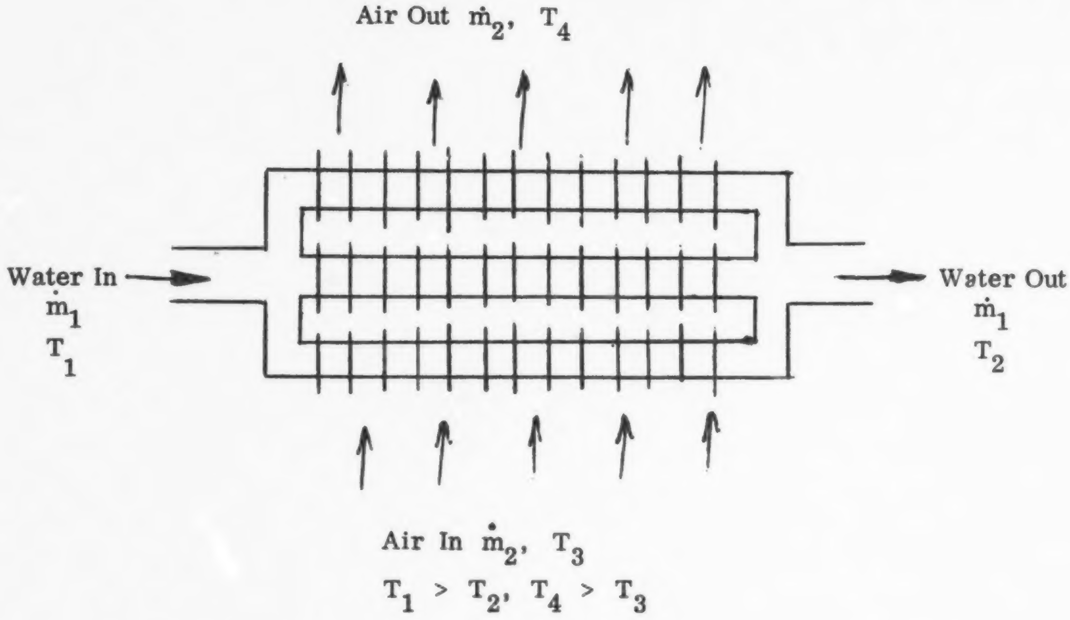


Figure 3.16. Cross Flow Air Heat Exchanger

The following relations are also obtained from heat balances upon the water and air, respectively

$$Q = \dot{m}_1 \ C_{p2} \ (T_1 - T_2) \qquad (3\text{-}23)$$

$$Q = \dot{m}_2 \ C_{p2} \ (T_4 - T_3) \qquad (3\text{-}24)$$

where $\dot{m}_1$ = mass flow rate of water

$\dot{m}_2$ = mass flow rate of air

C_{p1} = specific heat of water at constant pressure

C_{p2} = specific heat of air at constant pressure

Equations (3-22), (3-23), and (3-24) can be used in two basic ways: to solve for the heat transferred by a specified heat exchanger (UAF_G) given $\dot{m}_1$, $\dot{m}_2$, T_1 and T_3, or to determine a design which will transfer a given amount of heat (Q) for given values of the same variables. When the unit is forced draft, the air flow rate ($\dot{m}_2$) is known from the operating level of the fans. However, when a natural draft system is used, $\dot{m}_2$ depends upon T_4, T_3 and the airflow resistance of the heat exchanger and tower walls.

For a given plant, a value of Q_{rej} is known for each value of $\dot{m}_1$, and T_1 (T_c). From this, a heat exchanger and cooling tower can be designed to transfer the proper amount of heat for each value of T_1. To develop a system which will function properly under adverse conditions, the design is usually based upon high values of Q and T_1, and the highest expected value of T_3 for the geographic location of the plant.

The actual design calculations for designing a dry cooling tower, particularly a natural draft unit, are more involved than those given here. These equations are intended to provide an introduction to the concept. References 3.33, 3.34, 3.35, and 3.36 discuss in greater detail the engineering aspects of these heat exchangers.

3.6.4 Size Estimation

Equations of the type presented in Section 3.6.3 are used to estimate the size of the dry cooling tower required to reject a specified amount of heat to air at a given ambient temperature. In order that the plant function properly under most atmospheric conditions, a design air temperature near the highest expected value for that locality is used. The system can then be expected to perform effectively for all lower temperatures, since the tower will be able to reject more heat than required under those circumstances.

Because of the high cost of a dry cooling tower system compared to conventional cooling towers, particularly close attention is paid to the plant costs. It does not appear to be economical to replace the cooling equipment of an existing plant with a dry cooling tower because either the tower would have to be extremely large, or the plant would not operate at its design point. The best design requires an overall economic optimization study, as discussed further in Section 3.6.5.

There are only a few power plants in operation which use dry cooling towers, so it is difficult to predict what size cooling towers are necessary for given power requirements. However, the dry cooled system at Rugeley, England should serve to illustrate the expected component sizes for a medium size power plant. This particular plant generates 120 Mw using steam at 1,500 psig and 1,000°F reheated to 1,000°F between turbine stages. The hyperbolic cooling tower, built of reinforced concrete, is 350 feet high with a base diameter of 325 feet and a throat diameter of 205 feet. The heat exchanger has a total frontal area of 80,000 square feet and will reject 5.87×10^8 Btu/hr. The turbine operates at a vacuum of 28.7 inches Hg. The corresponding saturation temperature is about 80°F. A plant built to operate almost anywhere in the United States would require slightly higher exhaust pressure and temperature to avoid unreasonably large exchanger areas since our average yearly temperatures run higher than those in England.

3.6.5 Cost Information

A dry cooling tower is generally larger and more expensive than a conventional (wet) cooling tower. As a result, the design of a particular power plant should be preceded by an extensive economic optimization study. Although it is technically possible to build a dry cooling tower to produce turbine exhaust pressures as low as those obtained with water cooled condensers, this is usually not economically feasible. Instead, the system as a whole must be analyzed to determine the design which will result in the lowest cost per kilowatt generated, taking into account the added expenditures associated with decreased vacuum levels.

One optimization approach is to treat heat exchanger area and air throughflow rate as independent variables. Associated with these variables are condenser cost, tower cost, piping and construction cost, turbine price for various operating vacuum levels, pump and fan power costs, turbine power output at various vacuum levels, and the capital value of the plant power output. The procedure is to seek values of the independent variables which will minimize the power generation cost ($/kwh). An example of this technique has been presented by Oplatka (Ref. 3.37). For the particular cost criteria he selected, the variation of cost per kilowatt-hour as a function of condenser size is illustrated in Table 3. Note that the optimum condensing pressure is greater than that generally used for water cooled condensing plants.

Table 3

ECONOMIC OPTIMIZATION OF DRY COOLING TOWER SYSTEM

	Condenser Surface Area too Small	Optimum	Condenser Surface Area too Large
Relative Price of Condenser and Accessories	0.8	1.0	1.21
Pressure in Turbine Exhaust, in Hg	3.56	2.64	2.07
Difference in Power at Turbine Shaft Compared with Optimum, % of Rated Output	-2.35%	-----	+1.84%
Fan Power as % of Turbine Rating	1.6%	2.0%	2.4%
Difference in Cost of Generation Compared with Optimum Cents/kwh	+0.0018	-----	+0.0012

An optimization study of this type should be performed for all proposed power plant designs. However, such an analysis cannot be carried out for an arbitrary plant. The entire calculation is dependent upon the relative price of materials, labor and capitalized cost of power output, plus local temperature conditions for the specific location of interest. At the present time an insufficient number of plants utilizing dry cooling towers have been built to permit the determination of average costs. Oplatka has presented some relative cost data comparing river cooled, wet tower cooled and dry tower cooled plants (Ref. 3.38). These data are contained in Table 4.

An economic study is also necessary to best determine the location and cooling technique for a new power plant. When cooling water is available in large quantities at low cost, the dry cooling tower is not competitive with once-through cooling and conventional wet cooling towers. However, the dry cooling tower concept relaxes the requirement for an ample water supply. Land sites which are less expensive and closer to sources of fuel can now be considered. An entirely new approach is needed to determine the best site and appropriate cooling technique to be used.

Oplatka (Ref. 3.38) has developed a method for comparing three types of cooling for a particular plant: direct condenser cooling using river water, a water cooled condenser with a conventional cooling tower, and a "Heller System" direct contact spray condenser with a dry cooling tower (Oplatka refers to this as an air cooled condenser; we reserve this classification for conventional condensers rejecting heat directly to the air). Each system is required to generate the same amounts of power, 750 Mw. A conceptual plant is developed for each of the three cooling techniques, each design being optimized to produce the lowest cost per kilowatt-hour. Costs are then assigned to each plant including capital outlay and auxiliary power requiremements, but neglecting the cost of providing cooling water. From these data, one can determine ranges of water cost for which each system is desirable. The cost of water should include facilities for pumping, handling and treatment (e.g., construction and maintenance of a dam and reservoir). The proper type of plant design can thus be determined for a specific combination of location, water cost and required power output. By extending this type of study to a number of potential sites which may possess different land costs, cooling water costs and fuel availability, an economically optimum choice can be made. Other factors such as thermal or secondary pollution effects can be incorporated into the analysis by assigning additional costs related to the magnitude of the undesirable characteristic.

		I Cooling By River Water	II Wet Cooling Tower	III Dry Cooling Tower
1.	Pressure at Exhaust Bar. inches Hg	1.50	1.90	3.50
2.	Net Output of Turboset After Subtracting Power to Drive Feed Pump, Mw	750	750	750
3.	Power for Cooling Water Pump, Mw	1	3	-
4.	Power for Pumps and Fans in Dry Cooling Tower, Mw	-	-	12
5.	Efficiency	0.409	0.405	0.394
6.	Extra Capital Outlay Compared to I, 10^6 \$	-	2.48	3.07
7.	Capitalized Value of Power for Pumps and Fans Compared to I, 10^6 \$	-	0.16	0.83
8.	Capitalized Value of Efficiency Difference Compared to I, 10^6 \$	-	0.60	2.18
9.	Total Extra Capital Outlay Compared to I, 10^6 \$		3.23	6.09
10.	Total Extra Capital Outlay Compared to II, 10^6 \$		-	2.84

Table 4

COMPARISON OF THREE COOLING TECHNIQUES

3.6.6 Maintenance Requirements

Operational experience at Rugeley has shown that very little maintenance is required by a dry cooling tower system. Originally, it was feared that the circulating water would absorb aluminum and iron from the heat exchanger system, leading to boiler scale and corrosion of the heat exchanger and piping interior surfaces. After operation, however, internal corrosion of the heat exchanger was undetectable and the aluminum content of the cooling water was very low, of the order 0.02 parts per million. Therefore, water quality control for such a system is no more difficult than that required for a conventional water cooled condenser plant.

Exterior corrosion on the heat exchanger surface was found to be a problem of the Rugeley plant. This was attributed to atmospheric pollution, high humidity and a high concentration of salt in the air. Protective coatings for the heat exchanger surfaces are presently being developed which promise to eliminate the problem. External corrosion has not been a problem with dry cooling towers in less humid climates, such as Hungary, which implies that corrosion protection may only be necessary under particularly severe climatic conditions.

3.6.7 Secondary Pollutant Considerations

There is no evidence that the use of a dry cooling tower system leads to any significant secondary pollution. Since the cooling water flows in a closed loop through the plant and heat exchanger, the only external effect is to increase the temperature of the air passing across the heat exchanger surface. Fog and misting, which can be a problem with conventional cooling towers, cannot occur because there is no water evaporated into the air stream. Boiler blowdown can introduce contamination if dumped into a river, but this is associated with the boiler and has no relation to the cooling technique being used.

3.7 Evaporative Condensers

3.7.1 Introduction

The evaporative condenser, like the cooling tower, uses the latent heat of vaporization of water to achieve cooling. Existing units of this type have capacities ranging from 2 to 375 tons; no units of the capacity required by a power plant are known to exist at this time. Three hundred-seventy five tons would be sufficient to cool a 750 kw nuclear plant. They are generally used with industrial water coolers and air-conditioning systems, particularly in areas with water shortage problems.

3.7.2 General Description

Figure 3.17 is a diagram of an evaporative condenser. The substance to be cooled enters the coils in a vapor state, is condensed, and leaves the coils as liquid. Atmospheric air is pulled or blown by fans so that it flows over the outside surface of the coils. A continuous water spray is directed over the coils by a separate water circulation system. Heat is neither added nor taken from the circulating water, so it reaches an equilibrium temperature which depends upon the operating conditions. This temperature is always above the dew point of air, which causes the water wetting the coils to evaporate into the air stream absorbing the latent heat of vaporization from the coils (approximately 990 Btu/pound of evaporated water).

A big disadvantage with this arrangement is the fact that the ratio of vapor specific volume to liquid specific volume is several orders of magnitude greater for steam than for any of the common refrigerants. For this reason, steam is condensed on the outside of tubes in a condenser since very large diameter piping would be required for condensation on the inside. It is possible, however, that a power plant using a fluid other than steam may find an evaporative condenser attractive.

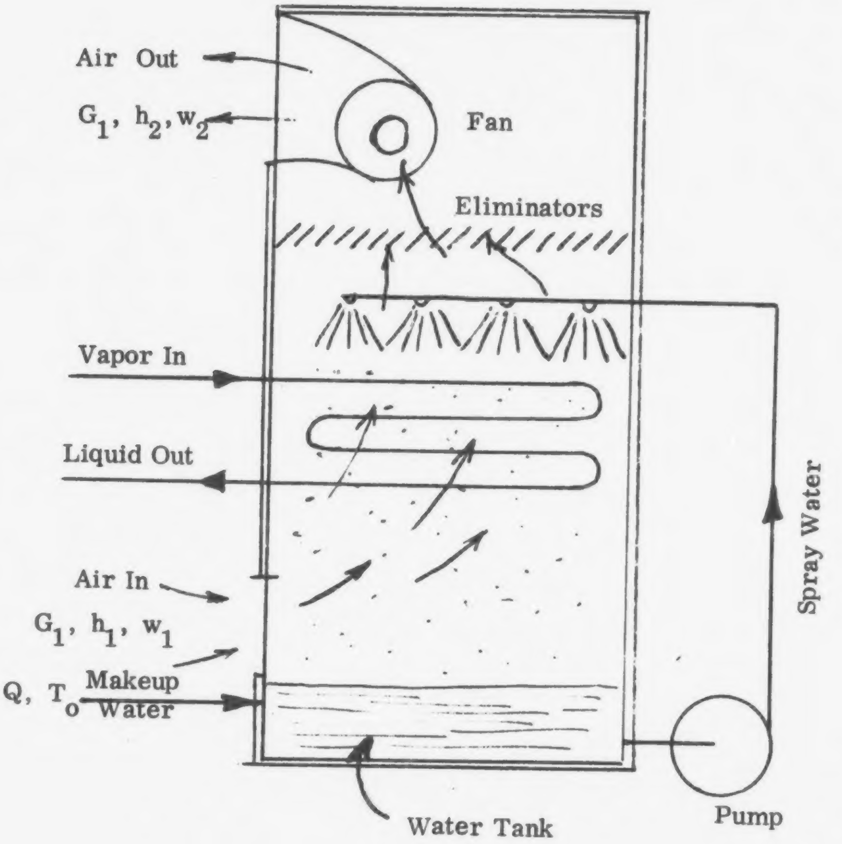


Figure 3.17. Schematic Diagram of an Evaporative Condenser

3.7.3 Theory of Operation

The heat flux transferred from the condensing refrigerant to the air is given by the following equation:

$$H = G\,(h_w - h_1)\left(1 - e^{-f_g A/0.24\,G}\right) \qquad (3\text{-}25)$$

where A = outside area of condenser tubes, (ft^2)

f_g = coefficient of heat transfer between air and water, (Btu/hr-ft^2- °F)

G = mass flow rate of air, (lb_m/hr)

h_w = enthalpy of air at a wet bulb temperature equal to the spray water temperature, (Btu/lb_m)

h_1 = enthalpy of entering air, (Btu/lb_m)

H = heat flux, (Btu/hr).

In order to use Equation (3-25), the enthalpy h_w must be determined, which requires a knowledge of the spray water temperature. The overall heat transfer coefficient from the refrigerant to the air (U) is given by the relation below:

$$\frac{1}{U} = \frac{1}{f_w} + \frac{B}{f_R} \qquad (3\text{-}26)$$

where B = ratio of outside to inside tube surface areas

f_R = heat transfer coefficient between refrigerant and tube wall, (Btu/hr-ft^2-°F)

f_w = heat transfer coefficient between water and tube wall, (Btu/hr-ft^2-°F).

By balancing the heat flux across the tube wall, and neglecting the thermal resistance of the tube material, the equation below is obtained.

$$\frac{T_R - T_w}{h_w - h_1} = \frac{G\left(1 - e^{-f_g A/0.24\,G}\right)}{UA} \qquad (3\text{-}27)$$

where T_R = condensing temperature of the refrigerant, (°F)

T_w = spray water temperature, (°F).

By creating a modified psychrometric chart, as illustrated in Figure 3.18, Equation (3-27) can be solved graphically. The angle θ is defined by

$$\tan\theta = \frac{G\left(1 - e^{-f_g A/0.24\,G}\right)}{UA}.$$

Point A in the figures is defined by the condensing temperature of the refrigerant (T_R) and the entering air wet bulb enthalpy (h_1). From point A, a line is drawn upward at an angle of θ from the vertical. The intersection of this line with the saturation line locates point C. The coordinates of point C are T_w and h_w. Thus, a knowledge of f_g, A, G, U, T_R, and T_1 is sufficient to determine the spray water temperature, and the air enthalpy at a wet bulb temperature equal to the spray water temperature. With h_w known, Equation (3-25) is used to calculate the heat flux from the refrigerant to the air.

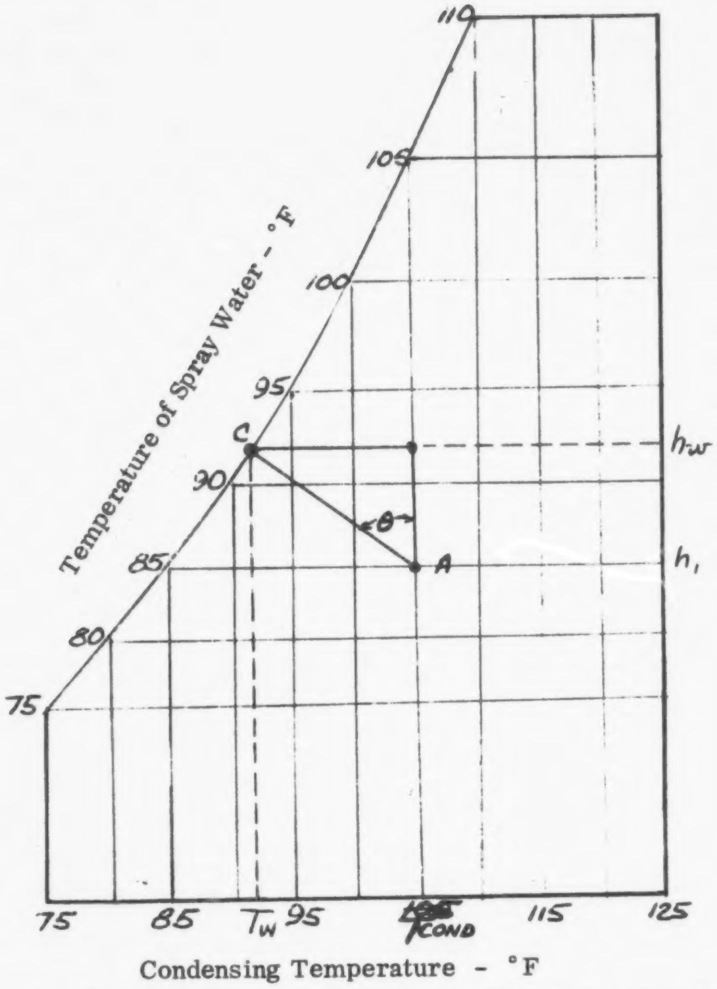


Figure 3.18. Modified Psychrometric Chart for Determining the Air Enthalpy at a Wet Bulb Temperature Equal to the Spray Water Temperature

3.7.4 Size Estimation

For a power plant, the heat rejected is known as a function of condensing temperature. Hence, the equations above are used to determine the condenser design parameters (e.g., A and G) so that the condenser will supply the proper condensing temperature and reject the required amount of heat. A discussion of heat transfer coefficients can be found in Reference 3.44. Judging from presently available equipment, the evaporative condenser is a very compact unit since it combines the function of a condenser and cooling tower.

3.7.5 Cost Information

Since evaporative condensers are not built in sizes large enough for even small power plants, no reliable cost information is available. However, Stoeker (Ref. 3.41) states that: "Cooling towers usually dominate installations larger than several hundred tons of refrigeration. In sizes below several hundred tons, the two compete vigorously."

With comparison to an air cooled condenser, Wile (Ref. 3.44) says: "The air-cooled condenser will have higher operating costs due to the higher operating pressure and generally larger fan motor as compared to the fan motor and pump of the evaporative condenser."

A very rough approximation to costs for very large units may be made by extrapolating manufacturers' costs of present day units. Such costs may be approximated as a straight line on a log-log plot of cost vs. unit size. The results are shown in Figure 3.19.

3.7.6 Maintenance Considerations

One of the principal problems in operating the evaporative condenser is the buildup of excessive salt concentrations in the circulated water. Martin (Ref. 3.42) recommends adding an additional quantity of makeup water on the order of one gallon per hour per ton into the spray system while subtracting a like amount from the sump.

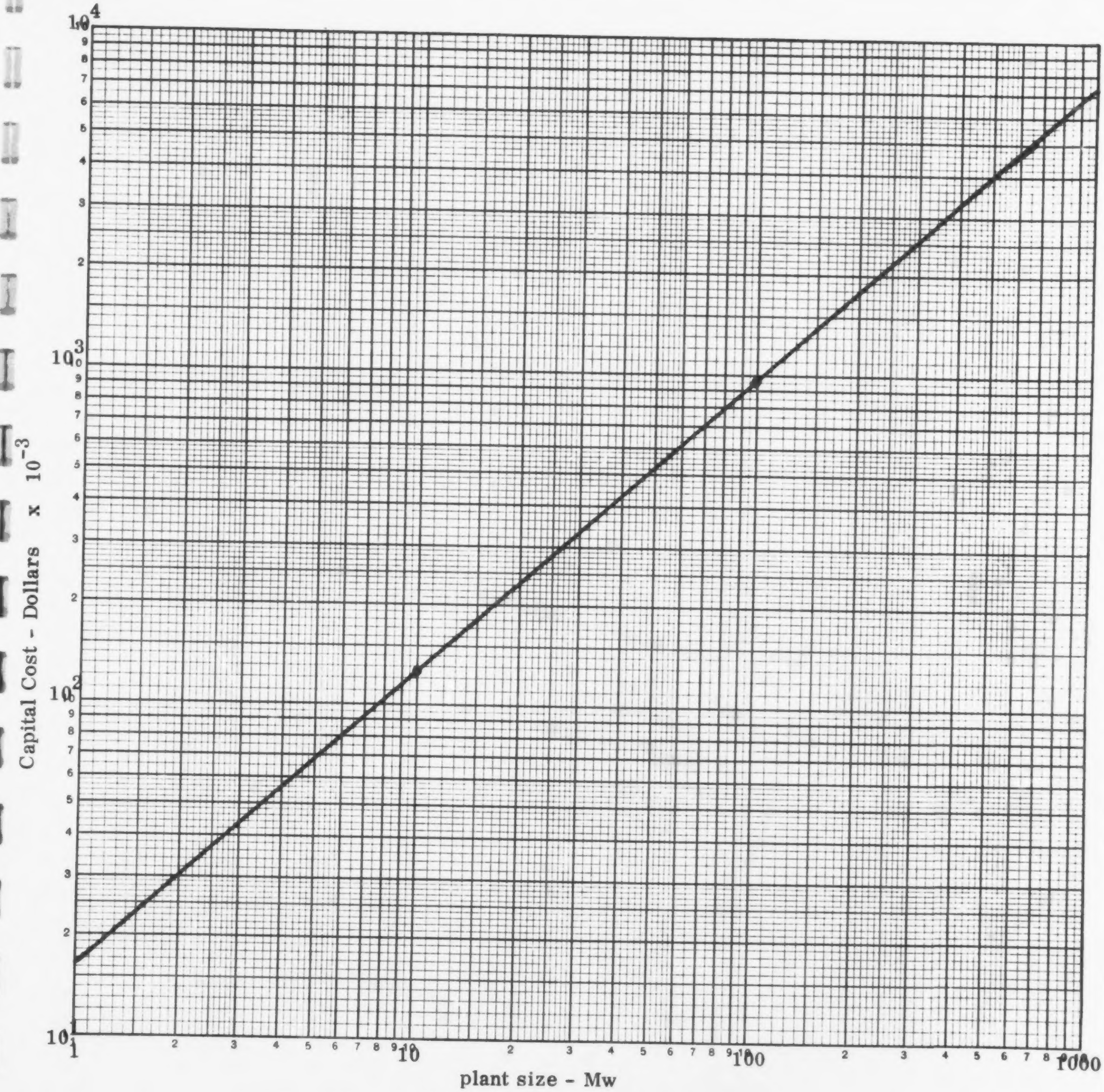


Figure 3.19. Capital Cost of Evaporative Condensers for Nuclear (η = 33%) Plants

A second problem is scale formation. It has a deleterious effect upon heat transfer ("fouling factor") and excessive amounts can affect proper distribution of the spray water. Observations in the field (Ref. 3.43) indicate, however, that evaporative condensers are less sensitive to scale formation than shell and tube condensers unless very heavy deposits are allowed to accumulate. The use of large water quantities is a simple method of reducing scale buildup and corrosion.

3.7.7 Secondary Pollution Considerations

There was no evidence of secondary pollution problems particular to the evaporative condenser reported in the literature. Ground fogging might be a problem in a very large unit since the height dimensions and air flow rates would be comparable to a mechanical draft cooling tower. The possibility of polluting a stream with the concentrated salt bleed water should also be considered.

3.8 Air Cooled Condensers

3.8.1 Introduction

The air cooled condenser rejects heat directly to air flowing over its exterior surface. Although this name is used by some authors when referring to dry cooling towers, we shall treat these as two separate classes of cooling devices.

3.8.2 General Description

A diagram of a basic air cooled condenser is presented in Figure 3.20. Steam flows through a long coil or matrix of tubes which have finned exterior surfaces. Heat transfer from the hot steam through the tube walls and fins to the cool outside air causes the steam to condense. In refrigeration service, the vapor usually enters the condenser in a superheated state, requiring that the condenser desuperheat the vapor as well as condense it. The steam entering the condenser of a steam power plant is usually a high quality binary mixture.

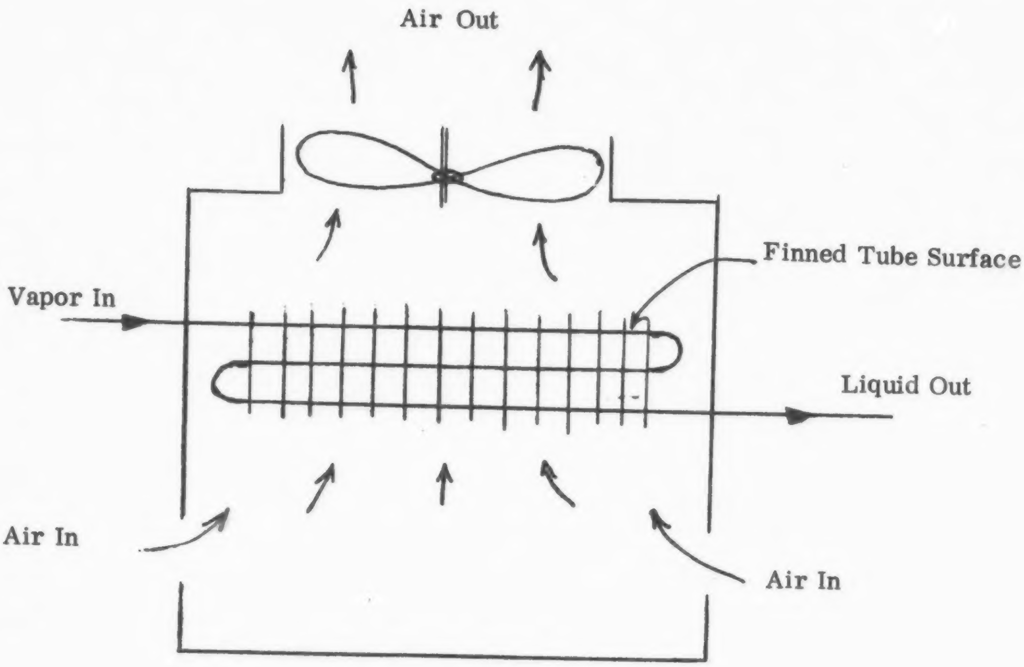


Figure 3.20. Schematic of Air Cooled Condenser

The heat transfer characteristics of an air cooled condenser are similar to the heat exchanger portion of a dry cooling tower; heat is transferred from the substance within the tubes to the exterior air stream. The basic difference is that vapor enters the condenser, is condensed, and possibly subcooled, whereas the heat exchanger portion of a dry cooling tower just cools a liquid within the tubes. Due to condensation, the air cooled condenser must transfer directly to the air the large amount of latent heat of vaporization (approximately 1,000 Btu/lbm, depending upon the condensing pressure) as well as the heat corresponding to the amount of liquid cooling.

Wet and dry cooling towers may be either forced or natural draft types. Because of the poor heat transfer characteristics of air, air cooled condensers nearly always use forced draft. The increased heat transfer rate to a fast moving air stream driven by fans permits size reductions which more than offset the cost of the air moving equipment.

It is difficult to obtain air cooled condenser information which is applicable to systems as large as thermal power generating plants. Most existing air cooled condensers are used with small refrigerating and air conditioning systems where compact size is a much more important design parameter than it would be for a power plant. A few large air cooled condensers are being used with power plants in Europe. One particularly successful unit is part of a 48 Mw power plant at Volkswagenwerke, Wolfsburg, Germany. A 3 Mw plant using an air cooled condenser is also in operation at Wyodak, Wyoming. Their results have been so successful that a 20 Mw air cooled system is being considered for that location.

3.8.3 Theory of Operation

An air cooled condenser may contain a series of three heat exchangers. The first acts as a desuperheater, lowering the steam temperature to the condensation temperature. This step would be rare in a steam power plant, however. The second is a condenser, transferring the latent heat of vaporization from a constant temperature liquid-vapor mixture between an initial state of pure vapor to a final state of pure liquid, and the final section acts as a subcooler, lowering the water temperature below the condensing temperature. The degree of desuperheating required depends upon the entering steam temperature, while the amount of subcooling depends upon the condenser size and the air temperature.

Each portion of the condenser transfers heat according to the heat exchanger equation as given below

$$Q = UAF_G \left\{ \frac{(T_1 - T_3)-(T_2 - T_4)}{\ln \left[\frac{T_1 - T_3}{T_2 - T_4} \right]} \right\} \quad (3\text{-}28)$$

where T_1 = water temperature entering heat exchanger section

T_2 = water temperature leaving heat exchanger section

T_3 = air temperature entering heat exchanger section

T_4 = air temperature leaving heat exchanger section

U = heat transfer coefficient

A = surface area of heat exchanger section

F_G = design section, experimentally determined for a particular heat exchanger configuration.

In the first section, T_1 is the entering steam temperature and T_2 is the condensing temperature corresponding to the operating pressure of the condenser. In the second section, $T_1 = T_2$ = condensing temperature, but Q is known from the heat of vaporization corresponding to the condenser pressure.

$$Q = \dot{m}_1 h_{fg} \quad (3\text{-}29)$$

where $\dot{m}_1$ is the mass flow rate of steam and h_{fg} is the enthalpy difference between saturated steam and saturated vapor. In the third section, T_1 is the condensing temperature. The theory of air-cooled heat exchangers is discussed in more detail in Section 3.6.3. Various design considerations and air cooled condensers can be obtained from numerous tests on heat transfer and fluid mechanics.

3.8.4 Size Estimatation

There are no standard techniques for sizing air cooled condensers since so few units of power plant size have been built. Data from small units cannot be extrapolated since they are designed for minimum size and usually operate at refrigerant condensing temperatures of approximately 105° F. Condensers for power plants operate at lower temperatures (70 - 80° F) for efficient system performance, and an economic design would be based on criteria other than minimum size.

The size of the air cooled condenser at Wyodak, Wyoming should serve as an example for a small power plant. The power output is rated at 3 Mw. The condenser is built in four sections, each section composed of a single layer of one inch diameter tubes thirty feet long. The tubes are spaced such that each layer is eight feet wide. The fins on each tube are 2-1/4 inches in diameter, giving a total exterior surface area of 101,048 square feet. Six fans, driven by 40 horsepower motors, develops an air flow of 3,180,000 lbm/hr. The cooling area of the condenser is approximately 35.0 square feet per kilowatt.

The plans for the 20 Mw unit are based upon the same ratio of cooling area per kilowatt. The turbine is expected to operate at an exhaust pressure of 6.0 inches of Hg. This value, higher than usually used for power plants, is more economical when the cost and effectiveness of the air cooled condenser are considered.

3.8.5 Cost Information

Except for the cooling system, the cost of components (turbine, boiler, etc.) will be about the same for plants using air cooled condensers or cooling towers. However, the capital cost of an air cooled condenser is approximately twice as much as a water cooled condenser and conventional cooling tower. This larger initial investment can be justified for applications where the air cooled plant operating costs are considerably lower. The operating cost of a conventional cooling tower (including pumping and water treatment) is approximately 0.35 mills/kw as opposed to a cost of 0.1 mill/kw for the additional fuel need when operating an air cooled condenser. This figure does not include the fan power, however, which is 1-1/2% of generator capacity. In addition, when the available water is of particularly poor quality, the water treatment cost for a cooling tower may be as much as 75% higher, as was the case at Wyodak.

Converse (Ref. 3.22) reports that an air-cooled condenser was considered for the Vernon, Vt. plant. Data from GEA Luftkuhlergesellschaft yielded an initial cost figure of \$15.6/kw (approximately 2x the cost for the cooling system with the ND tower). He also estimated electrical production would cost an additional 0.44 mills/kwh when the air-cooled condenser was used.

Other factors which enhance the desirability of air-cooled condensers are related to their low water requirement. The Wyodak plant, for example, is constructed at the mouth of a mine which produces sub-bituminous and lignite coal. Although this coal is very cheap to produce, the heating value per pound is so low that it is not economical to transport or store it. By producing power at the mine mouth, great savings in fuel costs can be realized.

As water costs and treatment requirements increase, the air cooled condenser system gains an edge over the river or cooling tower cooled plants. The water at mine mouth locations is generally very hard and acidic. Even when large amounts are available, treatment costs for a cooling tower become prohibitive. Thus, to utilize the low fuel cost of a mine mouth plant, air cooled condensers and dry cooling towers are particularly attractive.

3.8.6 Maintenance Requirements

Maintenance procedures for an air cooled condenser are primarily limited to the air moving equipment. Periodic oiling of fan motors, lubrication of bearings and adjustment of driver mechanisms is necessary. Occasional cleaning of the exterior finned surfaces to remove dirt and dust is suggested for efficient operation. Since the water flows in a closed loop through the boiler, water treatment is needed only to maintain a water quality which will not corrode the boiler or cause scaling. This degree of purification is sufficient to prevent corrosion and scaling of the interior surfaces of the condenser coils.

A wind deflector may be desirable when prevailing winds are likely to interfere with the discharge of air passing through the condenser system. Corrosion of the exterior surface will not be a problem if non-ferrous materials are used. When steel is required, a protective coating, such as those being developed for dry cooling tower heat exchangers, can be applied.

3.8.7 Secondary Pollution Considerations

The air cooled condenser introduces no secondary air or water pollution. The circulating air is heated as it passes over the condenser surface, but the humidity ratio is not changed. Noise generated by the fans and the air flow may be a nuisance in some locations although the sound level is about the same as that generated by a conventional mechanical draft cooling tower.

3.9 Advanced Concepts

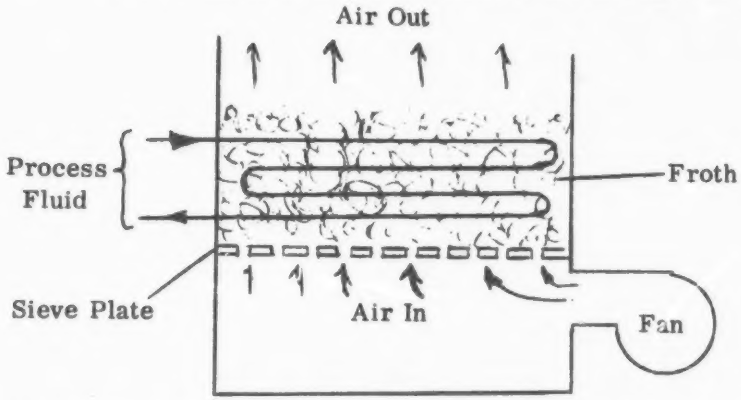
3.9.1 Introduction

There are several large-scale heat rejection schemes which may be classified as advanced concepts. Discussions have appeared in the open literature and, in some cases, models have been built and tested. None, however, has been utilized in a full scale plant so cost and operating information are not available.

Also included in this section is a discussion of some new materials for use in cooling towers. Strictly speaking, this is not an advanced concept but it is possible that new materials may reduce capital cost, construction cost, and/or maintenance costs. Such changes may alter the economic balance and favor an alternate heat rejection scheme.

3.9.2 The Froth Contact Heat Exchanger

A schematic of a froth-contact heat exchanger is shown below.



In this device, first described by Poll and Smith (Ref. 3.51), the process stream flows inside tubes which are submerged in a pool of water. Air is blown up into the water through the sieve plate creating a bubbly froth which covers the tubes. The process involves both latent and sensible heat transfer although the major portion is accomplished by latent transfer. Higher overall heat transfer coefficients than in conventional evaporative condensers have been observed because of the highly agitated liquid in close contact with the tubes and the high interfacial area between the air stream and the liquid in the froth.

Water requirements for a froth contact exchanger are comparable to other evaporative cooling devices. Slightly less may be necessary since preliminary tests indicate that a somewhat greater portion of the heat may be transferred as sensible heat. This is not yet definitely established, however. It is not necessary to circulate the froth water, but makeup to replace that lost by evaporation and to keep the level of dissolved solids and pH under control is still required. Much less air is required than for an equal capacity air-cooled condenser. According to Smith and Poll (Ref. 3.51), "The amount of air required by a froth-contact heat exchanger for a given load is, at a maximum, one-quarter of that required by the equivalent air-cooled unit and is normally much less."

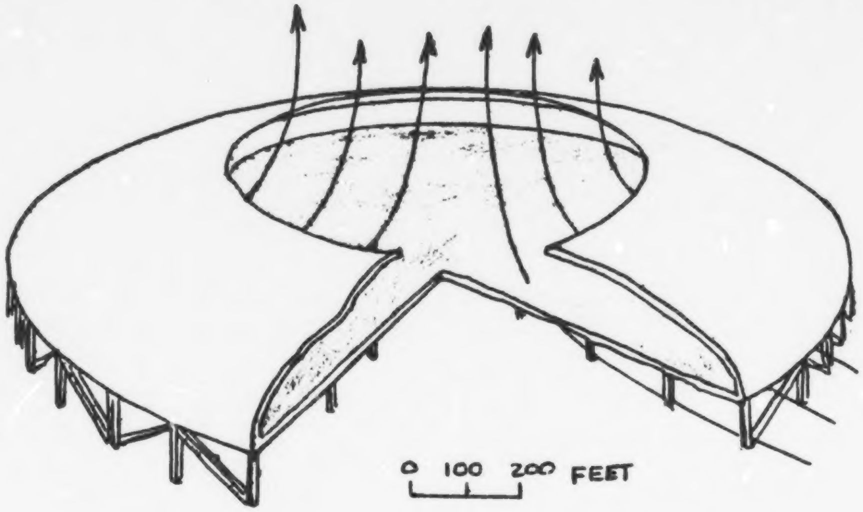
Several people in this country and in Britain are interested in the froth contact exchanger, primarily for use in refrigerant condenser service. However, due to the problems encountered when attempting to condense steam on the inside of tubes, a more likely function in a power plant would be replacement of the dry cooling tower in a "Heller system" (Section 3.6). These dry cooling towers are quite large and it is conceivable that a froth contact device could reduce the size appreciably with only a small increase in the amount of water required.

3.9.3 Ellipsoidal Cooling Tower

The following discussion is taken from Reference 3.52. To make cooling towers less conspicuous, the tendency has been to make them broader rather than taller. However, a point has been reached where performance may be adversely affected by wind. Under certain circumstances the wind flowing over the leeward rim of the tower may cause a down draft. This may happen in either wet or dry towers although it is more likely to occur in a dry tower because of the large diameter. Secondly, observations at Rugely have shown that winds adversely affect the heat transfer ability of the tower by causing poor airflow distributions.

To overcome this problem, engineers at Britain's Central Elecricity Research Laboratories are working on a new semi-ellipsoidal tower as shown on the following page.

This novel idea evolved from the knowledge that flow over an elliptic airfoil separates at a point about 60 to 70 percent of the chord from the leading edge. Model studies have shown this to be true for the ellipsoidal shape tower since the separation point is downstream of the exit. Downdrafts and warm air recirculation are thus eliminated and the models indicate that performance is likely to improve as a result of the small but significant suction produced by the flow over the shell.

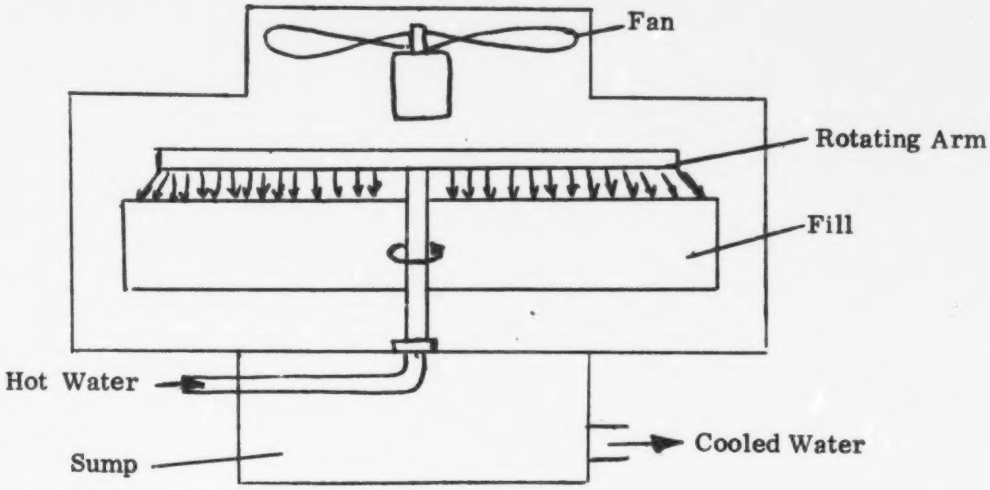


The design is applicable to wet or dry towers but present thinking indicates that the radiators of a dry tower could be arranged on a shallow inverted cone with its apex at the center of the tower base. Air would flow in and under the radiators as shown in the sketch above. It is estimated that the cooling capacity of a 300 foot high 1500 foot diameter dry tower would be sufficient for the needs of a 2000 Mw station. CERL engineers are testing a 1/40th scale, 36-ft-dia. wooden model using electrically heated radiators to test aerodynamic and thermodynamic efficiency of the design (Ref. 3.68). CERL emphasizes that this project is still in an early stage of development and many problems must be solved before a full scale installation is attempted.

3.9.4 Rotating Arm Cooling Tower

An unusual cooling tower design is described in Reference 3.53. A sketch of the unit is shown below. The force due to reaction of the curtain of water directed at the packing propels the distributor arm at 25 to 30 rpm. The plastic fortified wood-cellulose packing has numerous tiny vertical openings throughout its entire depth of only 18 inches. More space is saved by a special baffle attached to the trailing edge of each distributor arm only four inches above the fill which acts as a drift eliminator.

Because of these innovations the tower takes only about one quarter the space of more conventional cooling tower designs. It combines efficient cooling in the range of 120 to 500 tons (6×10^6 Btu/hr) with minimum weight and space.



The results obtained so far are encouraging; the improved heat and mass transfer result in smaller towers. Unfortunately, the cooling capacity of the units described is several orders of magnitude below that necessary for a moderate size power plant.

3.9.5 Rotating Cooling Tower

The following description is taken from Reference 3.53.

The rotary cooling tower has a revolving circular packing composed of plastic extrusions. Tip speed of the packing is about 1200 fpm. The packing has thousands of passageways traveling from a central core to the outside, following the involute of a circle. Water is sprayed onto the packing core from a stationary distributor running down the center of the core. Centrifugal force forces the water through the passageways as a thin film. As it leaves the outside of the revolving packing, the water strikes a honeycombed louvre which deflects it to the sump.

An induced-draft fan, mounted at the center of the core, draws air from the outside through the louvres and then through the passageways to the core where it is discharged to the atmosphere. The high velocity (30 fps) turbulent air stream flows countercurrent to the thin water film. The high relative velocity promises increased heat transfer rates and resultant decreases in size and weight.

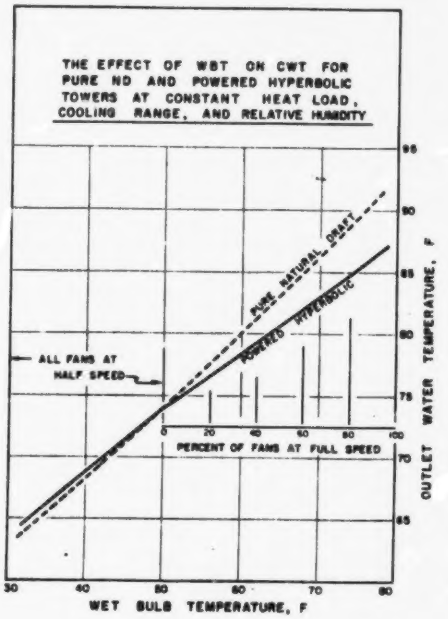
3.9.6 Powered Hyperbolic Towers

The powered hyperbolic tower has been suggested as a solution to the problems of unfavorable climate and need for better control associated with the ND hyperbolic tower (Refs. 3.69 and 3.70). The design (see Figure 3.21) is basically a natural tower with supplementary air moving capacity supplied by cells of axial flow fans mounted around the base. According to Reference 3.70 the capacity of an ND hyperbolic tower was doubled when fans were installed.

The benefits claimed for the design include:

1. More uniform outlet water temperature throughout the year.
2. Reduced ground area.
3. Improved control, enabling operation of associated equipment at higher efficiency.

The operating curve at constant heat load, range, and relative humidity is shown below.



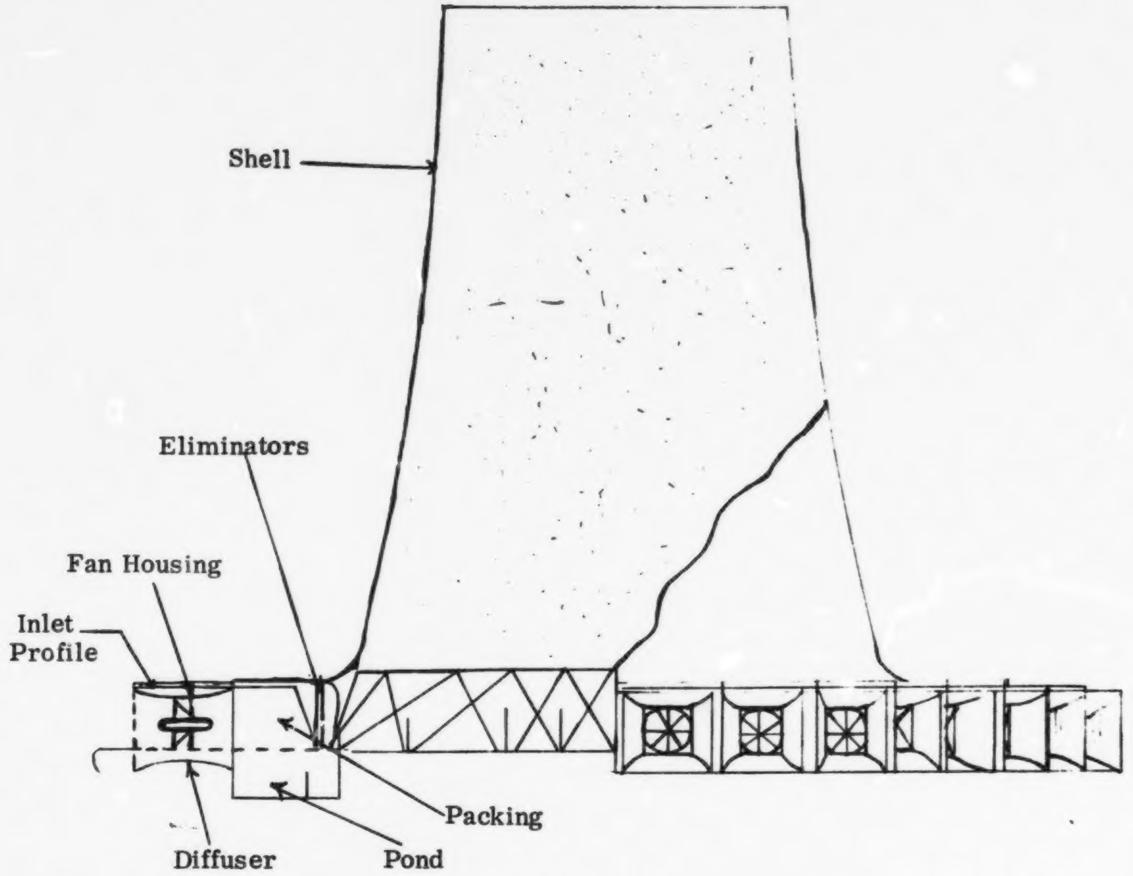


Figure 3.21. Sketch of Powered Hyperbolic Tower

3.9.7 New Materials

The utilization of new materials is aimed at improving the operational and maintenance side of the economic picture. Longer, trouble-free part life is obtained at slightly higher original cost. In the following paragraphs examples of the use of various modern materials are presented. The list is not intended to be exhaustive but merely to illustrate present day trends in material selection.

Big improvements have been made in fill materials. The high humidity, high temperatures, and concentrations of acids and salts contribute to rapid deterioration of the fill. Chemical treatment improves the service life of the standard redwood fill but presents the possibility of stream contamination. Hard-burned clay (acid-resistant) and high-impact plastics are being considered as alternatives for splash type packing.

In film type packing, where the water must wet the surface of the fill, several improvements are on the horizon. Plastic-fortified cellulose sheet which looks like corrugated cardboard is being tested. It is easily treated to improve fire resistance and to check growth of algae. Asbestos-cement sheets are already being extensively used. They have a rough absorbent surface that wets well. A stack of these sheets 5 feet high is equivalent to a 20 to 30 foot depth of splash packing. Minimum expected life is 15 to 20 years. Metal and plastics are also being formed into wavy sheets. A rather thick layer of water is formed in each of the hollows. Any solids precipitated during evaporation remain in the film and are washed down into the sump. Further water treatment may be necessary with this type.

Considerable effort has been made to cut down the noise generated by the fan. Some manufacturers are leaning toward squirrel-cage fans since they are usually quieter. However, they operate most efficiently in a specific speed range not commonly used in cooling towers, so power requirements may be high. Large multibladed fans using glass-reinforced plastic blades are also being designed. The favorable damping qualities of the plastic tends to eliminate high frequency vibrations that generate noise.

The exterior of almost all new cooling towers is composed of ceramic or concrete. Even smaller units are constructed in this manner. The durability of these materials combined with the scarcity of satisfactory lumber has been a deciding factor.

Fiberglass is gaining favor in several areas. Drift plates are now made of fiberglass; it allows greater flexibility in design. Fan stacks are also made of fiberglass. Fiberglass requires virtually no maintenance, no painting or treatment is necessary. A small all plastics cooling tower is described in Reference 3.54.

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Section 4

HEAT REJECTION EQUIPMENT - OPERATIONAL CONSIDERATIONS

4.1 Introduction

Off-design operation and other operational considerations affect both the capability and economics of heat rejection systems. Ambient conditions (temperature, relative humidity, amount of sunlight, wind, and rainfall) can alter the operating point of the system. Conditions within the system itself may change; water flow rate (or velocities), air flow rate or heat transfer coefficient may be a function of the operating point. Unusual problems which may affect the system performance are also discussed in this chapter. Clogging of nozzles, buildup of slime and algae on packing, and ice formation on inlets and fan blades all lower system effectiveness and result in higher operating costs.

4.2 Once-Through Cooling and Cooling Ponds

Changes in water flow rate, wind, and ambient conditions can alter the performance of these systems. A reduction in the flow rate raises the temperature of a river, for example, even though the upstream temperature remains constant since the water temperature rise at the discharge is given by

$$\text{temperature rise} = \frac{H}{\dot{m}\, C_p}$$

Thus, for a given H, the temperature rise increases as $\dot{m}$ decreases. Changes in river flow rate also effect the exponential temperature decay. A stream with a given overall exchange coefficient and equilibrium temperature will have both a greater average depth and mean velocity when passing a greater flow rate. This increases the value of α in the equation

$$T(x) = (T_o - T_E)\, e^{-\alpha x} + T_E$$

resulting in a decay to equilibrium temperature within a shorter distance.

The energy exchange coefficient (K) and equilibrium temperature (T_E) are functions of the meteorological conditions. The value of K increases as relative humidity, wind, and equilibrium temperature increase. The effect of these factors on the determination of the equilibrium temperature is more complicated and is dependent on several variables which are nonlinear functions of equilibrium temperature in addition to the energy exchange coefficient, air temperature, and net radiative input. The equations for T_E may be combined to yield

$$.051\, T_E^{2} + K\, T_E = H_R - 1801 + (K - 15.7) \left[\frac{0.26\, T_a + e_a + C(\beta)}{0.26 + \beta}\right]$$

The overall exchange coefficient is

$$K = 15.7 + (0.26 + \beta)\, bW.$$

The functions β and $C(\beta)$ can be represented by the following equations:

$$\beta = 1.965 \times 10^{-4}\, (T_E)^{1.90}$$

$$C(\beta) = 5.96 \times 10^{-6}\, (T_E)^{3.49}$$

It can be seen that while qualitative statements as to the effect of various parameters can be made, quantitative statements can be made only after numerical investigation.

In summary, it may be generally said that increased wind increases the effectiveness of rivers and ponds for cooling while increased sunlight, air temperature and relative humidity decrease their effectiveness.

4.3 Spray Ponds

Much of the discussion in the preceding section applies to spray ponds as well. In addition, the water flow rate, as determined by the pumping capacity, is a variable which may be affected by head (water depth), clogging of nozzles and pipelines, and deterioration of components of the pumping systems. Data is not available on, say, flow rate versus time in use. Obviously, a reduction in the flow rate will lower the effectiveness of the spray pond. A restriction in the nozzle or line will also lower the effectiveness by lowering the height of the water spray which decreases the droplet exposure time. This problem can be reduced through proper maintenance.

In extreme circumstances, icing may occur around the nozzles and on control valves. Depending on the circulation patterns within the pond, algae and slime can accumulate on various parts and cause fouling.

Increased winds may increase carryover substantially. Since the water droplets are pushed higher into the air, particles of water may fall into regions where they will not flow back into the pond. This problem is not likely to affect the capacity of a large pond, however.

4.4 Wet Cooling Towers

4.4.1 Off-Design Performance

In a given natural draft tower off-design operation may stem from changes in water loading or changes in ambient conditions. The available data seems to indicate that tower effectiveness is lower at reduced water loadings particularly in natural draft towers due to decreased air-water relative velocity. Mechanical draft towers have an advantage in this respect since they are able to maintain satisfactory air flow with reduced water loadings. If the heat exchange capacity falls off, the condenser back pressure will increase resulting in reduced plant capacity.

The following items may be considered ambient condition variables: wind, humidity, and wet and dry bulb temperatures.* Wind can have a considerable effect on range and heat rejected. Strong winds seem to adversely affect ND tower performance by creating poor airflow distribution. The effect is more pronounced on the windward side of the tower. Crawshaw (Ref. 4.1) found that the effect of wind was stronger at reduced water loadings. There is a small increase in draft but this is outweighed by the poor heat transfer due to the nonuniformities. If the pond is outside the tower and exposed to the winds, water loss will be increased, but the water will be cooler.

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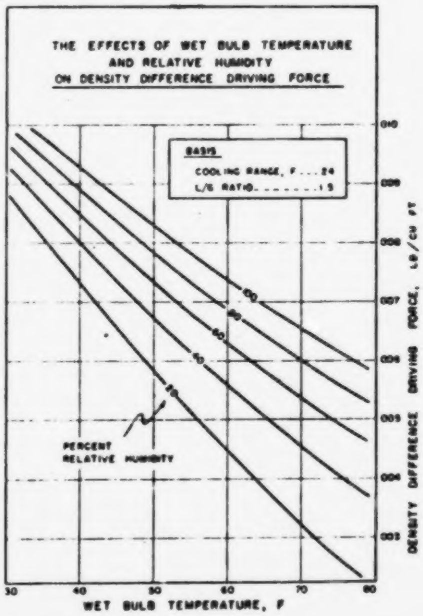
*Relative humidity is, of course, fixed once the wet and dry bulb temperatures are known.

Operation at a given wet-bulb temperature is more efficient when the relative humidity is higher due to the greater density difference. This is especially true in an ND tower. If ambient wet-bulb temperature, heat load, and water circulation rate are held constant for a given tower, the higher relative humidity results in cooler discharge water. The range will be the same, but the condenser pressure will be lower. Lower wet-bulb temperatures have the same effect. The driving force for heat transfer is greater (in both ND and MD towers) and the density difference is greater so that the ND towers operate more efficiently due to the higher velocities through the packing. With regard to this, it may occur that peak loads will occur in the summer due to high use of air conditioning, for example. This is an unfavorable situation for an ND tower since the wet-bulb temperatures are also highest then. The combined effects of relative humidity and wet-bulb temperature are shown in the figure below (taken from Ref. 4.7).



On very tall towers, appreciable vertical temperature gradients may be present. The variations inside the shell are of secondary importance. However, changes in temperature over the packing can produce considerable changes in frictional resistance and L/G ratios. Outside the tower, temperature inversions and vertical lapses reduce the range of the unit. Studies to date indicate barometric pressure has no measurable effect.

The frictional resistance of the tower is a function of the L/G ratio. Lowe and Christie (Ref. 4.8) found that the pressure drop across many of the packings doubled as the water loading went from 0 to 3000 lb/hr-ft^2 at low velocities (approximately 3 fps). The effect was less pronounced at higher velocities (7-8 fps). The spray above and below the packing also causes a pressure loss. To illustrate the effect of L/G ratio on the losses as the following data on a 238 high ND tower from Reference 4.9 is presented. The loss values are given in number of velocity heads:

L/G	1.0	2.0	3.0
packing	32.7	34.9	36.5
spray	3.8	9.6	14.8
eliminator	5.2	5.2	5.2
outlet	10.5	10.5	10.5
inlet	13.5	13.5	13.5
columns, etc.	2.3	2.3	2.3
Total	68.0	76.0	82.8

The packing resistance is due to skin friction over the water film and a form drag on the ripples observed on the sheets. The spray loss is assumed to be due to water droplets falling through the air stream above the packing and between the sheets. It is a very strong function of L/G. The last four losses are constant since the air flow was held constant.

4.4.2 Unusual Characteristics

Mechanical draft units seem to exhibit more problems than the natural draft units. Ground fogging has a greater tendency to occur with the mechanical units since they have much lower heights. The lower height also results in greater warm air recirculation which leads to problems of fan blade erosion and moisture condensation in gear boxes as well as reduced efficiency.

During cold weather operation, ice may form on inlets of both natural and mechanical draft units. The problem is more serious with the MD units since ice can actually break fan blades. Natural draft towers have been known to exhibit a mode of operation in which ice forms, is melted by the hot water, and then forms again. Such operation is not desirable and design precautions such as piping the hot water to regions around inlet should be taken.

Large natural draft towers do not have a history of producing fog or drizzle. Plumes are usually dissipated a few hundred feet from the tower and the great height keeps them well above ground level so recirculation is not a problem.

4.5 Dry Cooling Towers

4.5.1 Off-Design Operation

A dry cooling tower is designed to supply the required degree of cooling at the highest expected ambient air temperature. At lower temperatures the cooling rate is increased unless control techniques are applied. To maintain a fixed value of turbine exhaust pressure the airflow through the heat exchanger can be decreased by adjusting shutters in the tower, or by removing sections of the heat exchanger from service when large cooling reductions are required. Controlling the airflow is generally sufficient unless freezeup becomes imminent, at which point a reduction in the heat exchanger area is necessary. A cooling rate larger than design is not serious; in fact, it leads to a higher power output by decreasing the turbine exhaust pressure. However, for overall economic reasons, it may be best to maintain the vacuum level at the design value by controlling the volume of air passing through the tower.

When the air temperature rises above the design value, the turbine exhaust pressure and temperature increase until the temperature of the water in the tower heat exchanger is high enough that the tower can transfer to the air the amount of heat rejected in the condenser. The increased turbine exhaust pressure reduces the system efficiency. However, since the system is designed for the highest expected air temperature, the probability of occurrence of such an off-design operating condition is very small.

4.5.2 Operating Procedure

Freezing of the circulating water inside the heat exchanger during cold weather is a potential hazard with the dry cooling tower system. This is avoided by designing the heat exchanger in the form of separate units with control valves to put each individual unit out of service as desired. As the ambient air temperature decreases, the number of cooler segments actually in use is decreased by progressively removing more of them from service and draining them. At the Rugeley plant, an alarm is given when the water temperature drops to 45° F. The operator then reduces the effective surface area by actuating control valves which remove a heat exchanger section from use.

It has been shown that a particular plant should be designed to operate at an optimum turbine exhaust vacuum level. As the ambient air temperature changes, it is necessary to alter the cooling tower performance level to maintain the proper vacuum. Under extreme conditions, this is done by putting segments of the heat exchanger in or out of service as is done to prevent freezeup. However, for minor adjustments control is maintained by adjusting the air flow rate through the tower. Changing the fan speed will accomplish this in a mechanical draft unit. With natural draft towers a series of shutters within the tower can be positioned to vent the air flow to the outside of the tower before it has traveled through all the active heat transfer elements. Since the system is generally designed to operate properly at the highest expected ambient temperature, reducing the air flow through the heat exchanger provides a versatile method of controlling the turbine exhaust pressure.

4.5.3 Unusual characteristics

The most significant characteristic of the dry cooling tower system is its small water requirement. The paper "Dry Cooling Towers" (Ref. 4.4) presents the following estimated water requirements, which includes boiler make-up.

Once through water cooling	600 gpm/Mw
Conventional Cooling tower	30 gpm/Mw
Dry cooling tower	2 gpm/Mw

Clearly, the water requirement for a dry cooling tower is an order of magnitude less than that for a wet cooling tower, and several orders of magnitude less than that needed for a system using once-through cooling. Thus, a dry cooling tower is particularly attractive in locations where water is scarce or expensive. Less expensive land areas where water would have to be piped in for a conventional plant are particularly attractive locations. In arid regions, dry cooling tower or air-cooled condenser plants may be the only practical systems for power generation.

The winter operation of a dry cooling tower is very simple, requiring only that the proper number of cooling surfaces be disconnected to prevent freezeup. Since there is no moisture added to the circulating air, ice formation on fan blades or inside the tower is not a problem, as it is with a wet cooling tower which requires direct contact between the air and the cooling water.

Since the dry cooling tower requires a very small amount of make-up water, a minimum amount of water treatment is required. This is an added advantage over the conventional cooling tower, which requires thirty times as much make-up due to evaporation.

The "Heller System" does require controls not needed with a conventional cooling tower. Regulators must be installed which will close the spray valves if the circulating water pumps fail; otherwise, the condenser would flood. In addition, a throttle valve in the spray system is needed to prevent water hammer when the spray is shut off.

4.6 Evaporative Condensers

4.6.1 Off-Design Performance

The effects of changes in the wet-bulb temperature of the entering air and condensing temperature are illustrated in Figure 4.1. Clearly, the unit is able to handle higher loads as the wet-bulb temperature decreases. The capacity of the unit may also be increased by increasing the fan speed. The larger volume of air will raise the capacity but the extent depends on the refrigerant loading (tons per 100 sq. ft. of external area). Goodman (Ref. 4.1) indicates that, for a given U and f_g, the loading has a maximum value which is reached asymptotically as shown in Figure 4.2. Obviously, an increase in the air flow rate at operating point A will produce a larger gain in capacity than a corresponding increase at point B.

When a refrigerant is used, the vapor entering the condenser is sometimes superheated. This does not have a noticeable effect on the operation, however. The only adverse effect seems to be a slight increase in scale formations.

Present experimental evidence indicates that normal increases in water flow rate tend to increase the condenser rating. At high water flows, turbulence in the water-film and impinging spray increases the water film coefficient and the overall coefficient. The inside film coefficient remains reasonably constant.

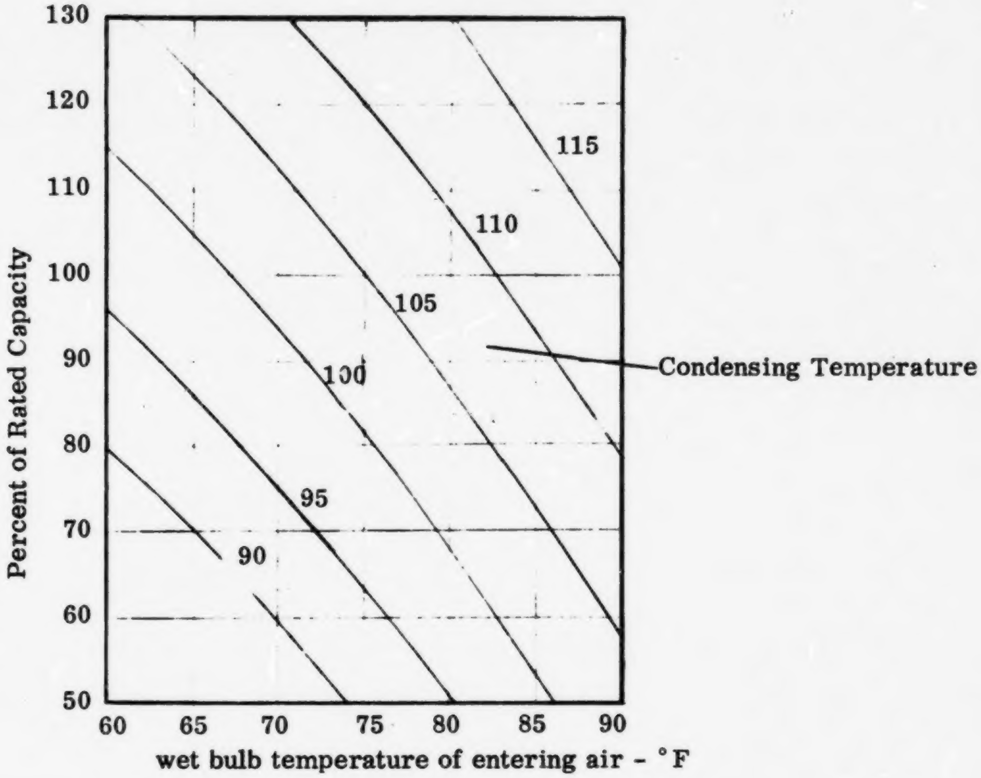


Figure 4.1. Performance Curves of an Evaporative Condenser (Condensing Temperatures Are For Refrigerant-12)

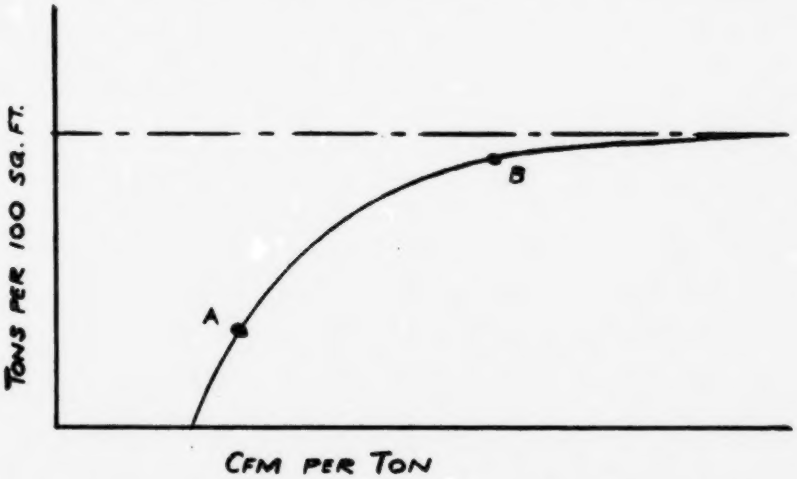


Figure 4.2. Evaporative Condenser Water Loading as a Function of Air Flow

4.6.2 Operational Procedures

For a very large plant, it is possible to combine several units to obtain the desired capacity. Water and vapor flow distribution problems tend to crop up, though, so care must be taken in controls design. At the present state it would require a very large number of these units to satisfy the cooling requirements of a large power plant (about two units per Mw).

During cold-weather, it might seem advisable to drain the water from an evaporative condenser and operate the coil dry. Figure 4.3 shows, however, that even at extremely low temperatures the dry coil capacity is below that possible with wet-coil operation.

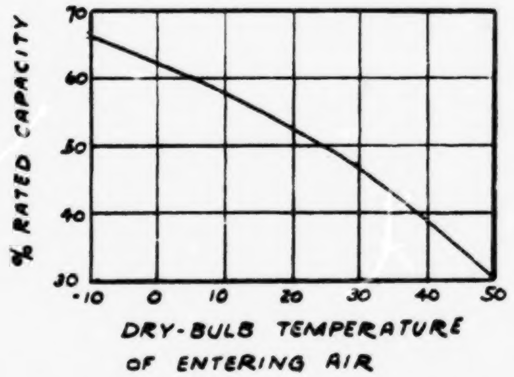


Figure 4.3. Capacity of an Evaporative Condenser Operating with a Dry Coil

4.6.3 Unusual Characteristics

Undesirable characteristics associated with evaporative condenser are generally associated with the fact that a refrigerant is being condensed in the tubes. For example, very warm days result in high gas discharge temperatures which tend to deteriorate the oil and lead to reduced compressor life. This, of course, would not be a problem in a conventional power plant where the condensate is returned to the feedwater pump.

4.7 Air Cooled Condensers

4.7.1 Operating Procedures

Except for periodic maintenance, the air cooled condenser requires no operational control procedures. Although freezeup may seem to be a possibility, no problems of this nature have occurred with systems presently in operation. Thermostatic controls which close air shutters or louvres as the ambient temperature decreases below freezing maintain the condensate temperature above the freezing point. The air cooled condenser plant at Wolfsburg, Germany has performed satisfactorily with ambient air temperatures ranging from - 7° F to 90° F.

4.7.2 Off-Design Operation

Since both the dry cooling tower and the air cooled condenser transfer heat to the air by conduction and convection heat transfer, they operate in basically the same manner under off-design conditions. The air flow can be reduced when the ambient temperature is low, or sections of the condenser can be taken out of operation under particularly cold conditions, usually when the air temperature is below freezing. Because most air cooled condensers are of the forced draft type, the air flow can be controlled by either changing the fan power or adjusting shutters and vents.

When the air temperature rises above the design value, the turbine exhaust temperature and pressure rise to new equilibrium values just as occurs with a dry cooling tower system. However, if the fans can be operated at a speed greater than design without damage, the cooling rate of the condenser can be increased during short periods of very high air temperature. Hence, it may be possible with some systems to maintain the proper system operating level by driving the forced draft fans at higher speeds.

4.7.3 Unusual Characteristics

The most significant characteristic of the air cooled condenser is its low water requirement; only boiler make-up is needed. As a result, it has many of the desirable characteristics of the dry cooling tower system. The economic aspects of a system which needs very little water is discussed in the section on dry cooling towers in Section 3.6.5.

Air leakage into the condenser can be a problem, since the condensing pressure is always well below atmospheric. Leaks of this type are much more difficult to detect than are leaks in the "Heller System", where water leaking outside the coils should be readily visible.

The biggest drawback to large scale air cooled condenser is the large piping required. Steam has a very small density (0.0053 lbm/ft^3 at 79° F and a pressure of 1.0 inches of Hg) compared to water (62.4 lbm/ft^3). To pass a given mass flow rate at similar velocities, the steam pipe would have to have an area approximately 10,000 times as large as the water pipe. Even though the steam will flow at a greater velocitiy for the same pumping power, the steam pipe will still have to be several orders of magnitude larger than that required for a water pipe. Attempting to use smaller steam pipes would require very high head pressures to move the steam, which would necessitate a large amount of pumping power. The "Heller System," dry cooling tower concept was developed from efforts to avoid this problem.

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Section 5

METHODS OF COMPARISON AND SELECTION

5.1 Introduction

When comparing the heat rejection schemes discussed in this report, several factors must be considered. Economics, unit size, local climate, and secondary pollution are among the more important considerations. Some categories, esthetics, for example, are less amenable to quantitative comparison than others making selection somewhat more difficult. Fortunately, the factors presently considered most important are easily defined and compared.

It is inevitable that certain assumptions be made in order to compare costs. Whenever this was done, care has been taken to list the assumptions clearly and, if possible, to estimate probable variations and their effect.

The systems discussed in Task I present various avenues of heat rejection to the atmosphere. None of the schemes makes an attempt at waste heat utilization which we feel is the ultimate solution so long as Rankine cycle steam plants are employed for power generation.

5.2 Economics

Data has been gathered on capital and operating costs for the various systems and has been presented in various sections of Chapters 3 and 4. This section is a summary of the data obtained. In some cases data on installations of the large size required for power plants were not available so estimates were obtained by extrapolation of the present data.

Two alterations to capital cost were made in this section. The cooling pond cost was set at $6.50/kw for fossil plants and $7.50/kw for the nuclear plants to reflect the greater amount of land necessary for the nuclear plant. The spray pond cost was raised to $8.10/kw for the nuclear plant since larger pumps are required.

A summary of the data collected in Task I is presented in Tables 5, 6, 7, and 8. The capital cost includes the price of the condenser, its associated pumps and piping, the heat rejection unit, and its accessories. The equipment cost was not given as the increase over once-through cooling because some of the schemes (particularly, dry cooling towers, evaporative condensers, and air-cooled condensers) use different condensing arrangements. The cooling system may be then thought of as a complete package which would be interfaced to the turbine exhaust and accomplish the condensation of the steam.

The operating costs listed in the tables are primarily power costs for pumps and fans and costs for makeup water and treatment water. Power requirements were established and converted to cost information assuming that electricity was available at 4 mill/kwh. Figures for water loss, usually as a percentage of water circulated, were used to determine power requirements for makeup water pumps.

The total cost column lists the equipment cost in dollars/kw-yr. The average annual fixed charge rate was assumed to be equal to 9%. The figure for a particular plant may be different since taxes and installation costs vary widely from place to place. The capital costs were amortized over a thirty year period to obtain the yearly figure.

Calculations were performed for both nuclear and fossil-fueled plants since the rates of heat rejection are quite different. The nuclear plant was assumed to have a thermal efficiency of 33% and an in-plant loss of 5%. The fossil-fueled plant was assumed to have a thermal efficiency of 40% and in-plant losses of 15%. The heat rejected to the cooling water is then:

nuclear plant:	6410 Btu/kwh
fossil-fueled plant:	3840 Btu/kwh

Calculations were performed for each plant allowing two temperature rises (10° F and 20° F) through the condenser. The following schemes have costs which are functions of the temperature rise of the cooling water through the condenser:

- once-through cooling
- cooling ponds

Cooling System	Equipment Capital Cost $/kw	Operating Cost $/kw-yr	Maintenance Cost $/kw-yr	Total Cost $/kw-yr	Additional Cost to Consumer %
Once Through Run-of-River	5.30	0.59	0.59	1.66	0
Once Through Estuary	6.30	0.59	0.59	1.75	0.05
Cooling Pond	6.50	0.74	0.74	2.06	0.23
Spray Pond	7.60	1.18	0.59	2.46	0.46
ND Wet Tower River Makeup	8.50	1.18	1.18	4.18	1.44
ND Wet Tower Reservoir Makeup	11.50	1.18	1.18	4.44	1.59
MD Wet Tower	7.20	1.54	1.54	4.43	1.58
ND Dry Cooling Tower	20.0	1.19	1.19	5.58	2.24
MD Dry Cooling Tower	13.0	1.54	1.54	5.65	2.28
Evaporative Condenser	10.0	1.05	1.05	3.70	1.17
Air Cooled Condenser	15.0	0.60	0.30	3.65	1.14

Table 5. Fossil Fueled Plant - $\Delta T = 10°F$

Cooling System	Equipment Capital Cost $/kw	Operating Cost $/kw-yr	Maintenance Cost $/kw-yr	Total Cost $/kw-yr	Additional Cost to Consumer %
Once Through Run-of-River	5.00	0.30	0.30	1.05	0
Once Through Estuary	6.00	0.30	0.30	1.14	0.05
Cooling Pond	6.50	0.38	0.38	1.34	0.16
Spray Pond	7.60	0.60	0.30	1.58	0.30
ND Wet Tower River Makeup	7.50	0.60	0.60	2.92	1.17
ND Wet Tower Reservoir Makeup	10.50	0.60	0.60	3.20	1.23
MD Wet Tower	7.20	0.94	0.94	3.23	1.25
ND Dry Cooling Tower	20.0	0.60	0.60	4.40	1.91
MD Dry Cooling Tower	13.0	0.94	0.94	4.45	1.94
Evaporative Condenser	10.0	1.05	1.05	3.70	1.52
Air Cooled Condenser	15.0	0.60	0.30	3.65	1.49

Table 6. Fossil Fueled Plant - $\Delta T = 20°F$

Cooling System	Equipment Capital Cost $/kw	Operating Cost $/kw-yr	Maintenance Cost $/kw-yr	Total Cost $/kw-yr	Additional Cost to Consumer %
Once Through Run-of-River	5.88	0.99	0.99	2.51	0
Once Through Estuary	6.88	0.99	0.99	2.60	0.05
Cooling Pond	7.50	1.24	1.24	3.16	0.37
Spray Pond	8.10	1.98	0.99	3.70	0.67
ND Wet Tower River Makeup	12.50	1.98	1.98	6.16	2.08
ND Wet Tower Reservoir Makeup	15.00	1.98	1.98	6.38	2.21
MD Wet Tower	9.40	2.34	2.34	6.23	2.12
ND Dry Cooling Tower	22.0	2.00	2.00	7.38	2.78
MD Dry Cooling Tower	15.0	2.34	2.34	7.43	2.82
Evaporative Condenser	13.0	1.40	1.40	4.67	1.34
Air Cooled Condenser	17.0	1.00	0.50	4.43	1.10

Table 7. Nuclear Fueled Plant - ΔT = 10°F

Cooling System	Equipment Capital Cost $/kw	Operating Cost $/kw-yr	Maintenance Cost $/kw-yr	Total Cost $/kw-yr	Additional Cost to Consumer %
Once Through Run-of-River	5.24	0.50	0.50	1.47	0
Once Through Estuary	6.24	0.50	0.50	1.56	0.05
Cooling Pond	7.50	0.62	0.62	1.92	0.26
Spray Pond	8.10	1.00	0.50	2.23	0.43
ND Wet Tower River Makeup	11.50	1.00	1.00	4.08	1.49
ND Wet Tower Reservoir Makeup	14.00	1.00	1.00	4.31	1.62
MD Wet Tower	9.40	1.33	1.33	4.20	1.56
ND Dry Cooling Tower	22.0	1.00	1.00	5.38	2.24
MD Dry Cooling Tower	15.0	1.33	1.33	5.41	2.26
Evaporative Condenser	13.0	1.40	1.40	4.67	1.83
Air Cooled Condenser	17.0	1.00	0.50	4.43	1.70

Table 8. Nuclear Fueled Plant - $\Delta T = 20°F$

- spray ponds
- wet and dry cooling towers

Operating costs are, of course, lower when the temperature rise is higher because less circulating water is required. Maintenance costs are also lower since the pumps, etc., are smaller and replacement parts are generally less expensive and easier to replace.

Evaporative condensers and air-cooled condensers remove essentially the latent heat of the condensing steam so the size is specified for a plant operating at a given condensing pressure. The unit size for a nuclear plant will be greater, of course, due to the larger amount of heat rejected. Performance does vary with changing ambient conditions, but these calculations are more extensive and will be performed in Phase II.

The total cost of some of the systems also includes the cost of running at higher condenser temperatures. The percentage increase in power production was calculated from the equation given by Converse.

$$\% \text{ increase} = 0.2\ (T_d - T)$$

where T_d = design condenser temperature

T = actual condenser temperature

T_d was assumed to be 100°F (1.93 inches Hg) for once-through cooling. The following condenser temperatures were assumed for the other systems:

System	T - °F	p - "Hg	Power Increase-%
Cooling Pond Spray Pond	100	1.93	0
ND Wet Cooling Tower	115	2.99	3.0
MD Wet Cooling Tower	110	2.60	2.0
Dry Cooling Tower	120	3.45	4.0
Evaporative Condenser	110	2.60	2.0
Air Cooled Condenser	120	3.45	4.0

A figure of 0.42 gpm/kw for circulating water rate for fossil fueled plants has been used in several places in the report. This corresponds to a plant with a thermal efficiency of 40% and a temperature rise through the condenser of 18°F. In this section ΔT's of 10°F and 20°F were assumed. The circulating water rates are found to be:

ΔT	Fuel	gpm/kw
10	f	0.76
20	f	0.38
10	N	1.28
20	N	0.64

Since reliable information on maintenance costs was not available, the maintenance cost was set equal to the power cost following the recommendation of Lane and Larson except for spray pond and air cooled condensers. It was felt that this method yielded figures which were excessive for these units.

The total cost column contains some interesting results. The costs of cooling are much higher for the nuclear plants. Secondly, the power penalty attached to some of the systems is evidently quite substantial. For example, in Table 6 the capital costs for the spray pond and the ND wet tower with river makeup are approximately equal. The operating and maintenace costs are also about the same. However, the 3% power penalty attached to the tower results in 0.6% additional cost to the consumer. It is possible that a power penalty should be attached to the spray and cooling ponds as well.

There is also quite a difference between the towers which require reservoirs to be built and those which have natural water available for makeup and purge duty. The additional cost is about $0.25/ kw-yr. The dry towers do not require such reservoirs but initial cost is very high. The air-cooled condenser is more expensive than the wet towers but may prove to be the most economical choice when water is scarce. The evaporative condenser is really an unknown quantity and, of course, makeup and purge water are still necessary.

The last column in Tables 5, 6, and 7 lists the additional cost of the cooling system to the consumer. Generally the cost to the consumer decreases as the range is increased except for the evaporative and air-cooled condensers which increase relative to the other systems as the range is increased.

5.3 Size and Capacity

There is considerable variation in the amount of land required by the various cooling schemes. However, a direct comparison of the cooling schemes might be somewhat misleading if one is not familiar with power plant design. For example, one would expect a considerable saving in space required when an evaporative condenser was compared to a mechanical draft wet cooling tower. It is common practice to place the condenser directly under the turbine-generator set. Installation costs would probably be higher for the MD wet tower scheme but the evaporative condenser would probably be larger since the condensation occurs on the inside of the tubes so the two units might have equal land requirements.

Generally, we can say that the size is inversely proportional to the heat transfer ability of the unit. Improvements such as spraying the cooling water or increasing the velocity of the airstream will result in decreased unit size. Units which use only air for cooling are much less efficient than evaporative units.

It is possible that the last three schemes will not be practical on very large stations due to the large amount of space required. Air-cooled condensers are a particular problem since they have both the problem of poor outside heat transfer (tube to air) even with fans and the problem of trying to condense steam on the inside of the tubes. Perhaps a binary cycle using a fluid on the low-end that has a small ratio of specific volumes is a possible solution.

5.4 Geographic Limitations

Several geographic considerations have been mentioned in the preceding sections. Extremes in temperature, relative humidity and other climatic conditions such as excessive rainfall can affect the cooling systems. The location of a plant can have a considerable effect on the economics. Plants located at mine-mouth, for example, can usually use more costly cooling systems but can still be competitive due to lower fuel costs. Systems which use very little water are necessary there since the water requires a large amount of treatment.

The availability of water has obvious economic implications. Runnoff may be necessary to maintain the level of a cooling pond or to provide a supply for cooling tower blowdown or makeup. The quality of the water is also important since water treatment is expensive both in terms of dollars per pound of chemicals and increased maintenance costs.

The topography of the region surrounding the plant may influence the operation of the cooling system. A particular arrangement of mountains could cause winds and air circulation patterns that force vapor plumes from cooling towers to remain around the plant. Poor draft patterns could also result. High surrounding mountains could raise the cost of fuel transportation.

The most important climatic factors are temperature (wet-and dry-bulb), amount of rainfall, and wind. One must remember that evaporative cooling devices can "approach" the wet bulb temperature of the air while the dry devices can cool only to the air dry bulb temperature. This means that a dry cooling device would be unable to maintain proper condensing pressure during a very hot summer day. On the other hand there are no ice formation problems in winter months.

The effect of rainfall on the operation of the various systems has not been studied. At this point we feel that this should not present a problem in any part of the United States. The amount of wind has a negligible effect on mechanical draft units. In natural draft units the wind does increase the draft somewhat but tends to produce nonuniform air distributions which lower the heat transfer ability.

5.5 Availability

The following cooling systems described in Sections 3 and 4 have been used in large power stations (500 - 1000 Mw) in the United States:

- once-through cooling
- cooling ponds
- ND and MD wet cooling towers

Smaller stations have utilized MD wet cooling towers, spray ponds, and air-cooled condensers. No one in the United States has used the dry cooling tower or evaporative condenser concept in a power plant. In all of the most recent studies of situations where once-through cooling was not practical, the ND wet cooling tower was found to be the most economical. However, they still require large amounts of makeup and purge water (about 1×10^6 gal/hr for a 1000 Mw plant) so that if it becomes necessary to site plants in areas where water is not plentiful or requires much treatment, more intensive studies will be made of the dry units.

Section 6

SUMMARY OF TASK I

6.1 Present Status

At this stage of the investigation a complete documentation of all available techniques and equipment for rejecting large quantities of heat either to waterways or to the atmosphere is in hand. Information is available whereby estimates may be made of equipment size, capital cost, and operating cost, both in terms of maintenance dollars and parasitic power drain. These estimates, however, are restricted primarily to so-called conventional operating conditions, and are based on standard design procedures.

This information is useful primarily to provide baseline results relevant to the simplest kind of solution to the thermal pollution problem. That is, these results may be used to estimate the incremental operational and cost penalties involved in going beyond run-of-river cooling.

6.2 Relationship With Other Tasks

The output from Task II, Phase I, will be similar to that of this task, but will deal directly with the power cycles. The results of the two tasks can then be combined to map out a "state-of-the-art" operating region with associated cost factors for power production and related thermal controls under "ordinary" conditions; that is, for the plant operating at conventional temperatures and pressures, utilizing conventional cooling requirement of standard design.

In Task IV, Phase I, these results will be used as a starting point to determine what the desirable characteristics of a total-community power generation system would be. These results will also serve as the base data to Task I, Phase II, in which a model of the power generation process will be constructed, parametized, and optimization criteria developed.

The remaining tasks will deal with substantial advances in departures from the state-of-the-art when operating conditions, now considered far off-design, will be investigated. The present results will serve not only as a starting point for design modifications but also as a basis for comparision with existing cost and operational complexity.